POWER

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THE following is a message to the fireman:

Successful firing simply means the production of the largest amount of steam at a uniform pressure with the least amount of fuel and labor.

To be a successful fireman, a man must use intelligence. He cannot follow any fixed set of rules, do as his grandfather did or as he has been taught. He must study the plant conditions, the weather, load and fuel. What did nicely yesterday may not answer at all today. What produced good results in a former plant may not work at all well in the present one. Methods which the other fellow finds successful may not suit your own case even a little bit.

Whether to carry thick fires or thin; the matter of draft regulation; the amount of forcing necessary; the selection of the proper time to slice or clean fires; the size, kind and quality of coal to be used, all these should be left to the man on the job and they may be if he is of the right kind.

The man who is continually working at his fires, poking, leveling and stirring, cannot produce economical results.

First, study the conditions and then learn how to fire to suit those conditions. Stoke coal onto one side of the fire at a time if possible; stoke lightly and do not smother the fire; watch the results. Use your head more and the firing tools less; the use of the coal scoop is a fine art.

Mr. Fireman, you are at work at a trade. If you make good you are the man wanted; if not, some other man must take your place.

Remember, a good fireman is a valuable man in any steam plant. In the fire room you can lay the foundation for becoming a successful engineer or qualifying for other responsible positions. Make the most of your chances by carefully studying the needs of your work.

More boilers, arches and grates are ruined by poor management than by actual service. The fireman who first has an extremely hot and then a dead fire; first, the dampers wide open, then tightly closed; water up, then down; who shovels the coal in any old way, pokes, digs and levels the fires continually and then brags about how much coal he can handle, need not reasonably expect to advance or even hold his job for long.

The mechanical stoker is designed to do automatically and at the proper time what the human stoker sometimes neglects. The man may sit and read until his fires burn down, so that when he does get ready to work he has to rush the fires so as to get back what he wilfully lost. In other words, he uselessly burns up his employer's money. At least, the automatic stoker has not this failing. Oftentimes, the very man who is thus neglectful complains the loudest for not being paid more or advanced.

Remodeled Substation at Reading

With the completion of the new plant of the Metropolitan Electric Company at Reading, the old generating station, containing a great variety of apparatus, was converted into the main substation of the system. This necessitated completely remodeling the buildings and substituting entirely new equipment, at the same time keeping the station in operation.

This substation was briefly mentioned in connection with the description of the new power house which appeared in the November 22, 1910, issue.

At the time of the construction of the new power plant at the outskirts of the city, the old plant consisted of three buildings on Seventh street, in the heart of the city, containing the engines and generators, switchboards and regulators, with a large building in the rear, extending to Lemon street, which contained the boilers. A plan of these is shown in Fig. 1. Building No. 1, the first to be erected, was equipped with six small horizontal, high-speed engines located on the first floor, each belted to two small dynamos on the second floor. There was also a line of shafting on the first floor driving two rows of dynamos and arc machines on the second floor, this line shafting being driven by a 700-horsepower Hamilton-Corliss engine, located in an adjoining structure.

At this stage the equipment totalled fourteen 60-kilowatt, 125-volt Edison bipolar machines, eighteen arc dynamos, six small engines and one Corliss engine, supplying the Edison three-wire system and arc circuits. Switchboards for the entire Edison three-wire system and generators, series arc system and series incandescent system were also located on the second floor.

In an extension at the rear a 350-horsepower Green Corliss engine was added, belted through a jack-shaft to a 2300-volt, 60-cycle, two-phase alternator. The switchboard for this unit was located near the end of the jack-shaft, this being the first move toward high tension.

In 1894 building No. 2 was erected, in which were installed three vertical cross-compound engines, direct connected to railway generators of a total capacity of 1600 kilowatts, a vertical cross-compound engine direct connected to two 100-kilowatt, 125-volt direct-connected machines, and the railway switchboard.

Further growth was represented by the erection of building No. 3, containing at first two Ide engines, one belted to a Short railway generator and a Westinghouse 2300-volt, 60-cycle, 2-phase generator, the other belted to a Jenney multipolar generator and a Thompson-Houston railway generator. Later, there were installed three direct-connected units, an Allis-Chalmers cross-compound engine driving a 500-kilowatt railway generator and two Harrisburg cross-compound engines, one coupled direct to a 350-kilowatt, 2400-volt Stanley alternator, the other to two 300-kilowatt, 125-volt direct-connected generators.

Other additions were, three boosters and a railway battery, and two 400-kilowatt, 25-cycle Stanley rotary converters, supplying current to two 300-kilowatt

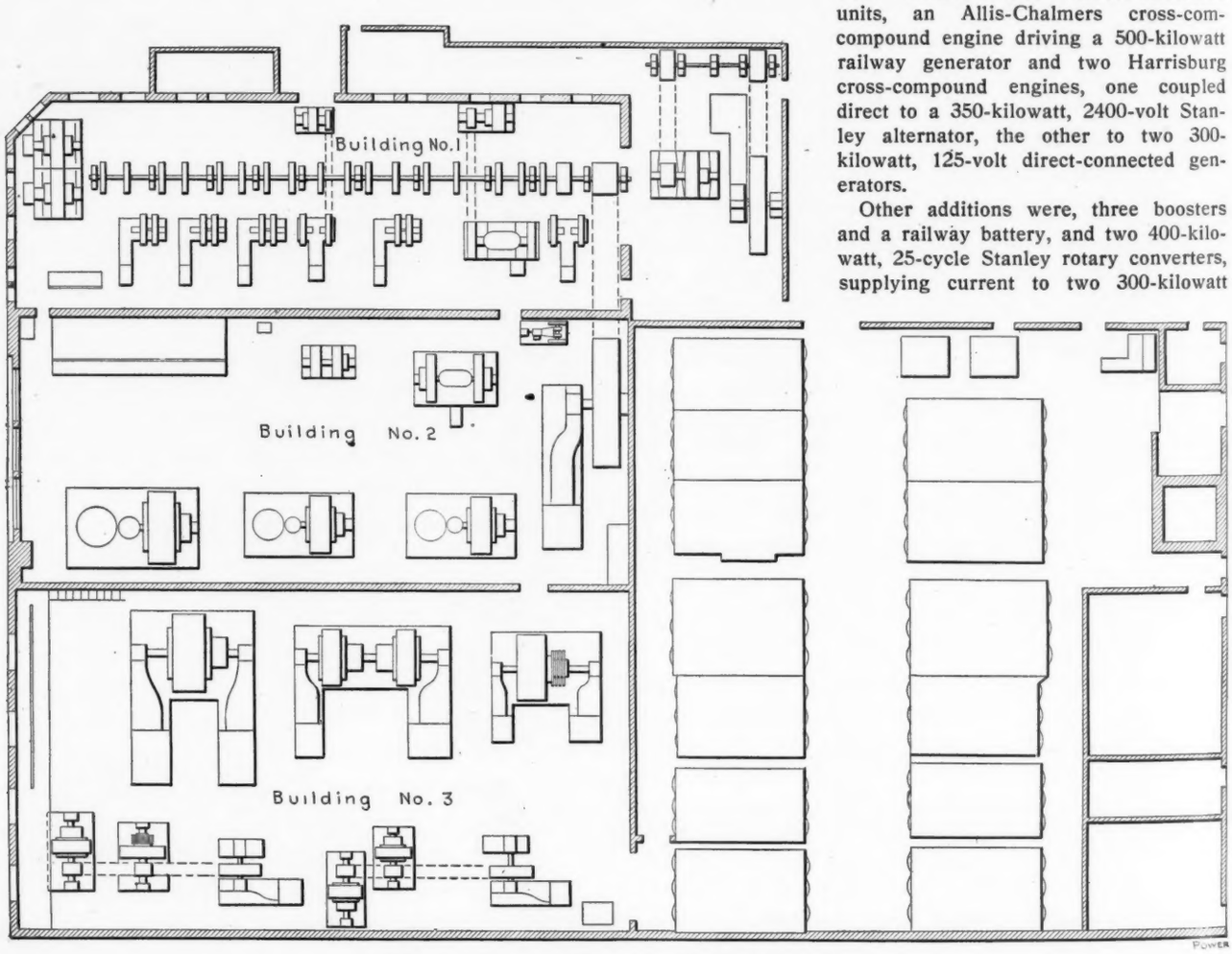


FIG. 1. PLAN OF BUILDINGS BEFORE REMODELING

substations through step-up transformers and a 15,000-volt transmission line.

The final rated capacity of this plant was 7800 boiler horsepower, and 5000-kilowatt generator capacity. Figs. 2, 3, 4 and 5 are views of this old equipment.

Reconstruction for Use as a Substation

When the new power house was built last year it was decided to convert the old plant into a substation. The first undertaking was the remodeling of building No. 1 without interrupting the operation of the apparatus. It was possible to put in a single row of steel columns down the middle, which would support steel floor beams laid above the old wooden floors, so that the old floors would serve as forms for the new concrete floors. The machines were raised to the new level one at a time; the belts were lengthened and the belt holes left in the new concrete floor to be closed up after the removal of all the old apparatus. The third floor being built, furnished protection to the apparatus so that the old wooden roof could be removed, and a concrete slab roof supported by steel trusses erected.

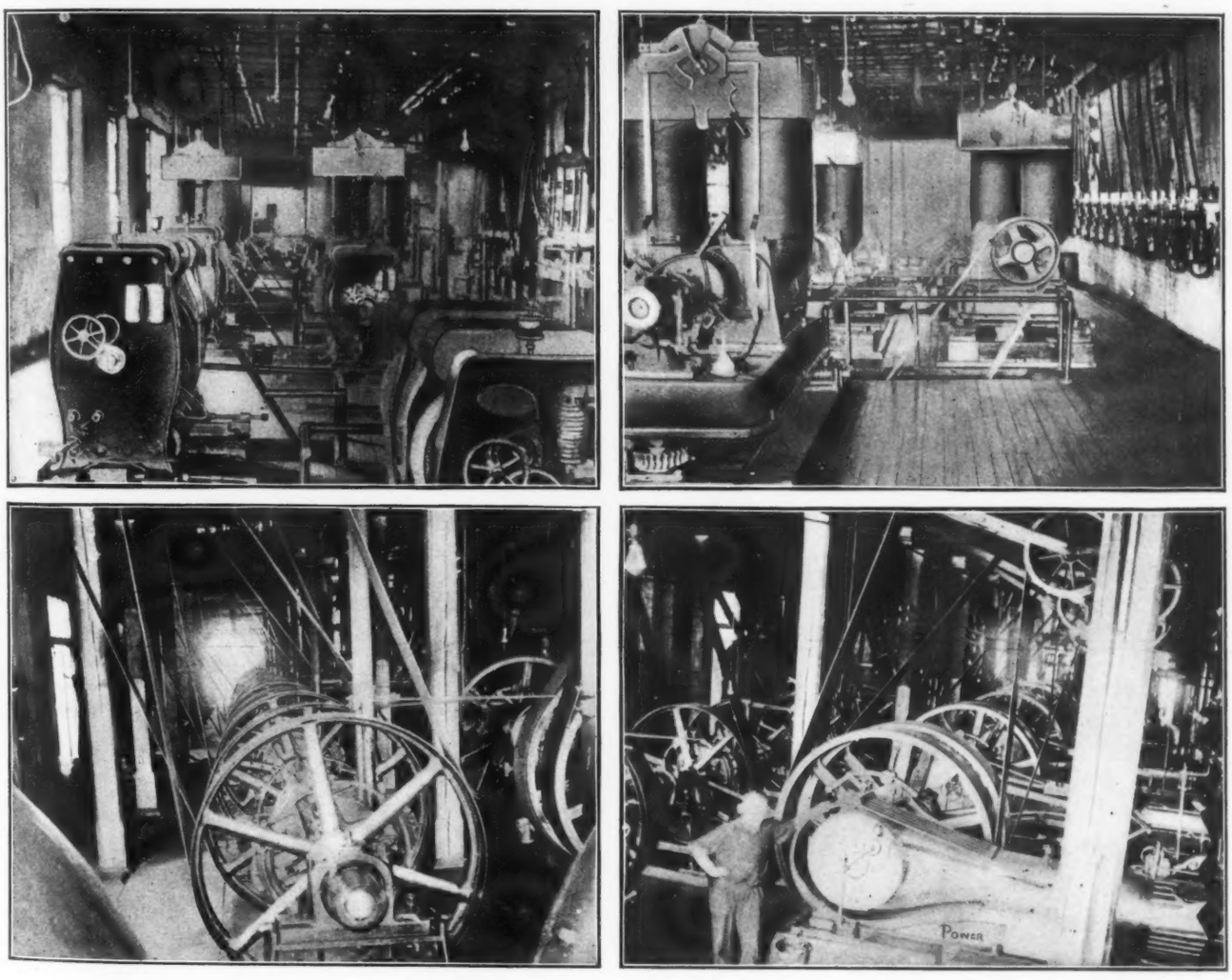
The structures on the side and rear end, which contained, respectively, the step-up transformers and the Green-Corliss engine, were torn down, giving more yard room for the pole lines, and improving the appearance of the property.

To give space necessary for the new equipment in No. 2 building, an addition was made, extending into the old boiler house, the rear end wall being moved back. The rearranged plant has 50 per cent. greater capacity, and occupies only 41 per cent. of the ground space taken up by the original plant. The general arrangement of the remodeled plant is shown in Fig. 7.

In both No. 1 and No. 2 buildings, the ground floors were practically a solid mass of concrete foundations for the engines, shafting and generators which had been installed from time to time. Along the walls, however, there were trenches for steam pipes which, with a slight amount of additional work, could be used for the air tunnels over which the air-blast transformers were placed. In order to install the fiber conduits connecting the transformers and rotaries to the high-tension switching systems, it was necessary to raise the floor levels of both buildings sufficiently to permit these conduits to be laid on the top of the old engine-room floor, and this resulted in the switch cell compartment floor being higher than the rest of the floor in No. 1 building, as shown in Fig. 9.

New Installations

The substation receives high-tension, three-phase, 60-cycle currents at 13,200 volts, through underground cables from the new generating station. Provision is made for connection to six of these cables and six high-tension switches were accordingly installed, four for present use and two for future needs. Current is distributed by this substation for railway feeders at 600 volts; for an Edison three-wire system at 125 and 250 volts; power circuits at 60 cycles, three-phase, 2300 volts; to lighting circuits at 2300 volts, 60 cycles, single



Figs. 2, 3, 4 and 5, Showing Old Equipment

phase; and to series mercury arc and series tungsten street lighting systems.

The rated capacity with the present installation is 8000 kilowatts, and provision is made for a future installation amounting to 3000 kilowatts, making the ultimate rated capacity 11,000 kilowatts.

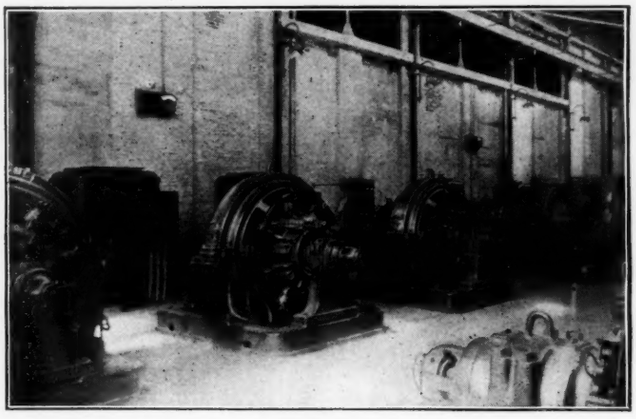


FIG. 6. RAILWAY ROTARY CONVERTERS

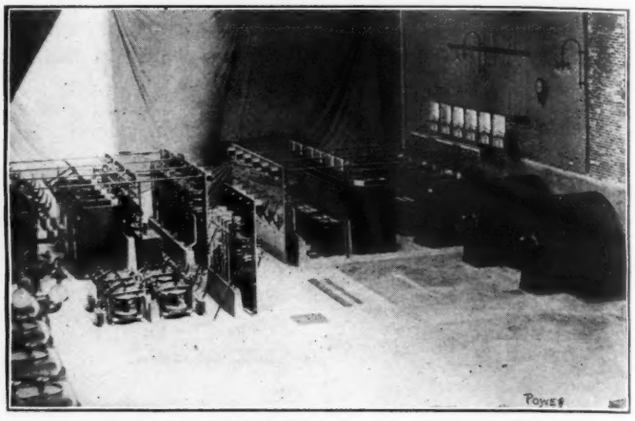


FIG. 8. ARC-CONTROL SYSTEM

BUILDING NO. 1

As rearranged (see Fig. 9), the first floor of the building contains the high-tension switches and busbars, office of the engineer in charge, and the arc lamp and meter repair department. The second floor contains the 2300-volt switches, and all of the switchboards except those used for the street-lighting system. The benchboard is located in front of an opening in the wall, which gives the operator a full view of the rotary room floor. The benchboard controls all of the high-tension switches which are located on the first floor, and the 2300-volt switches located on the second floor for the control of the low-tension side of the light and power transformers. The switchboard for the 2300-volt light and power systems is a vertical board controlling all the outgoing 2300-volt lighting and power circuits. These switches are remote-control solenoid-operated, and are arranged in two banks, over each bank being located a set of three busbars. Normally, one busbar set is to be used for three-phase power feeders, and the other for the single-phase lighting feeders, connected to separate transformers, so that the irregular motor loads cannot affect the lighting. In case of trouble, however, either set of busbars can carry either the lighting or power feeders, or both, and any transformer can supply either set of busbars, or both. The switchboard for the Edison three-wire system is composed of rotary converter control panels and feeder panels. The same arrangement is carried out in the railway switchboard. These three boards are arranged in one line facing the benchboard (see Fig. 10), so that everything on these boards is in full view and within easy reach of the operator.

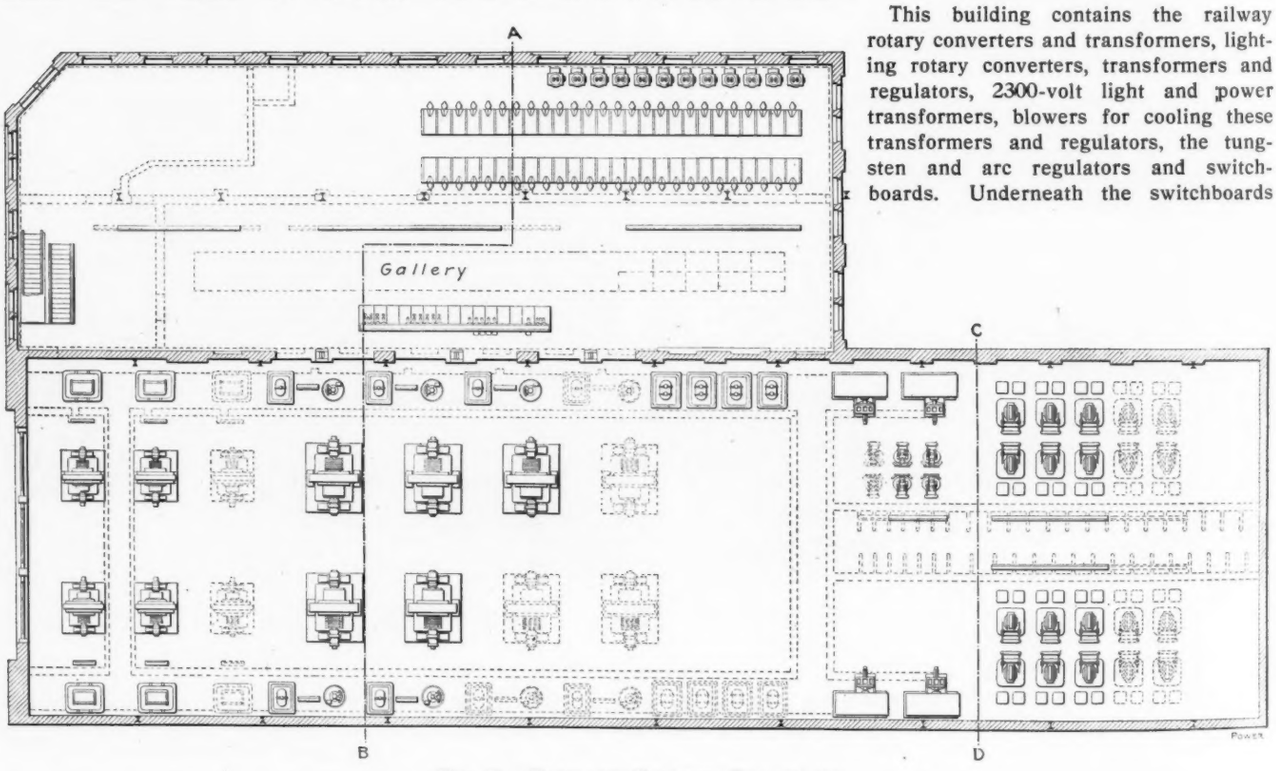


FIG. 7. PLAN OF PRESENT SUBSTATION

BUILDING NO. 2

This building contains the railway rotary converters and transformers, lighting rotary converters, transformers and regulators, 2300-volt light and power transformers, blowers for cooling these transformers and regulators, the tungsten and arc regulators and switchboards. Underneath the switchboards for the arc and tungsten systems is a tunnel which contains the 13,200-volt busbars supplying these systems, and in compartments are the oil switches, which are operated by rods carried up through

the floor to the back of the panels. This is shown in Fig. 10.

High-tension Switches and Busbar System

The high-tension switches are connected to a main busbar which is in three sections, and an auxiliary busbar which is in two sections. By this means, and the use of two busbar section switches, it is possible to isolate any line which is carrying a badly fluctuating load, and, if necessary, this line may be isolated back to the main busbar in the generating station, or still further, by means of the auxiliary busbar to one generating unit in the main station.

Two auxiliary switches are provided, which will, by means of the auxiliary busbar, replace any regular switch which may be out of commission, thus doing away with the necessity of providing duplicate feeder and transformer switches. This saved space and first cost.

Outgoing Lines

At present the railway, lighting and power feeders are taken out of the side of building No. 1, and the street lighting circuits out of the Lemon street end of building No. 2, all overhead.

The railway ground enters with the high-tension underground conduits at the front of building No. 1, where it is connected to a busbar in a manhole under the floor, the negative of each railway rotary converter being connected to the ground busbar.

The neutral for the Edison three-wire system is connected to a busbar in the tunnel. From here, separate cables are connected to the low-tension side of the lighting rotary transformers by means of switches on the starting panels. Only the positive and negative leads are carried to the distributing switchboard on the second floor.

All the rotary converters are started from the alternating-current side, through low-voltage taps, without reactances. They are provided with speed governors, those supplying the lighting circuits being regulated on the alternating-current side by six-phase induction type, air-blast, motor-operated regulators, controlled from the switchboard gallery.

The series mercury arc and tungsten circuits are regulated by air-cooled constant-current transformers, in which the movable coils are balanced by weights.

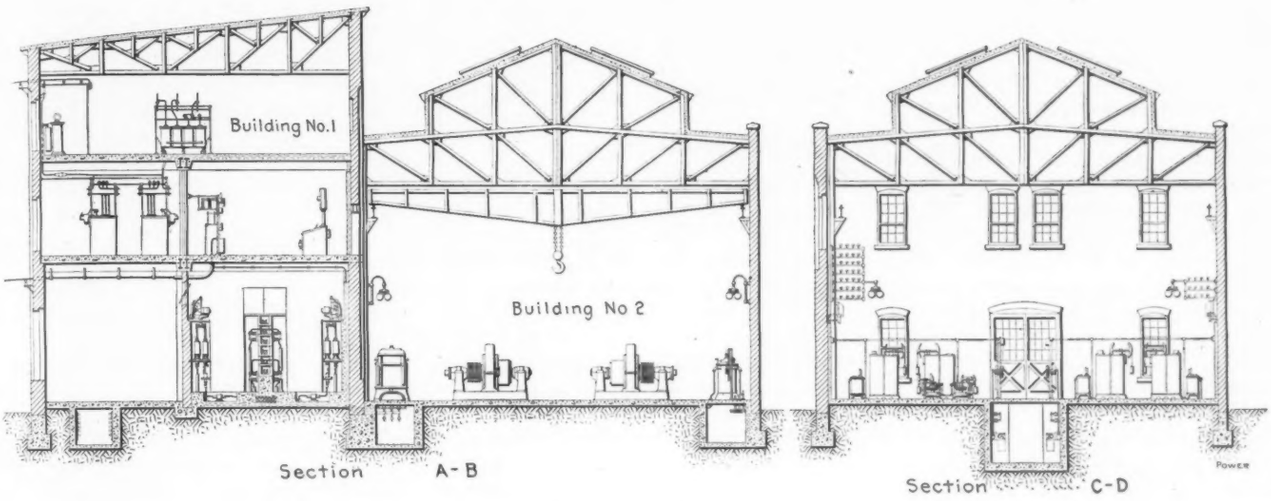


Fig. 9. Section through Substation Showing New Arrangement

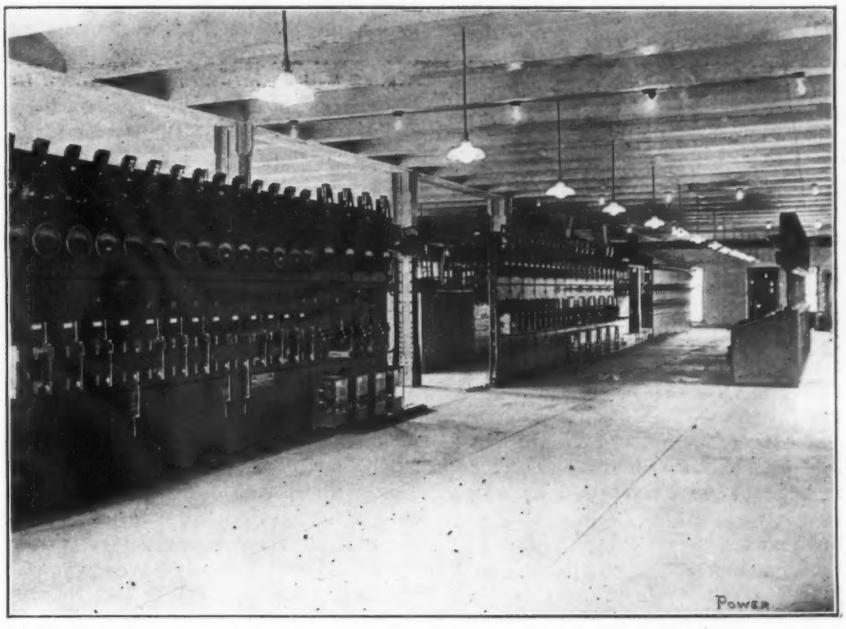


Fig. 10. Control Switchboard in Gallery

The design of the electrical equipment and the architectural and structural changes was done by Walter J. Jones, consulting engineer, of New York City, who also designed the main generating station.

Test of Large Boilers

The large double Stirling boilers at the Delray station of the Edison Company at Detroit have been tested, and it is said that the results are very interesting. The boilers were described in the issue of October 11 of last year. Each contains upward of 23,000 square feet of heating surface, and some 480 square feet of grate surface. The particulars of the test are not available for publication, as the Babcock & Wilcox Company, the makers of the boiler, are reserving them for presentation to a future meeting of the American Society of Mechanical Engineers, but it is understood among Detroit engineers that 5500 horsepower was obtained from a single unit.

Draft and Differential Gages

By Julian C. Smallwood*

The various types of draft and differential gages in general use are described and their salient features pointed out.

*Associate professor of experimental engineering, Syracuse University.

There are three units commonly used in the measurement of fluid pressure: pounds per square inch, inches of mercury and inches or feet of water. Steam pressure is measured in pounds per square inch, back pressure or vacuum in inches of mercury, draft in inches of water and water pressures in pounds per square inch or in feet of head. As a rule, the pressure of the atmosphere (14.7 pounds per square inch) is selected as the point from which measurements are made, the ordinary pressure gage reading so many pounds above atmospheric pressure, the vacuum gage so many inches of mercury below it and the draft gage inches of water above or below the atmosphere. In some cases merely the difference between two pressures is required, for which purpose a differential gage is used.

The conception of pressure in pounds to the square inch needs no comment. A cubic inch of water at 60 degrees Fahrenheit weighs about 0.036 pound; hence, a column of water 1 inch high and 1 square inch in cross-section will exert a pressure by its weight of 0.036 pound distributed over the square inch of its bottom surface of contact. If the column is made 2 inches high, this pressure is doubled, and so on.

Mercury, at 60 degrees Fahrenheit, weighs nearly 13.6 times as much as water; therefore each inch of mercury exerts a pressure of

$$13.6 \times 0.036 = 0.49 \text{ pound per square inch}$$

The simplest form of draft gage, the U-tube, is that shown in Fig. 1. This is a piece of glass tube bent in the form of a U and partly filled with water. It is connected to the flue at *A* by a rubber tube, and when the two water levels are the same the pressure in the flue is the same as that without. But if the level on the flue side of the U-tube is *H* inches higher than the other, then the pressure of the atmosphere on the open side of the tube is enough greater than the flue pressure to raise the weight of *H* inches of water against it. From this difference in level the difference in pressure is obtained between that in the flue and atmospheric pressure expressed in inches of water.

Such an instrument is easily made and can be put together at small cost. It is not always easy to bend the glass into the form of a U, in which case two straight pieces of glass may be joined by a piece of rubber tubing, as shown in Fig. 2. These may be mounted on a board to which a paper scale is glued and afterward varnished.

The simple U-tube is applicable also to pressures greater as well as less than atmospheric, the high level then being on the open side. For measuring pressures of more than one pound, however, the water gage becomes inconveniently long. Hence, for this purpose a heavier liquid, such as mercury, which will not rise so high, is used. Each inch of mercury being equivalent to 0.49 pound per square inch, indicates a pressure 13.6 times as great as that shown by water.

The scales of these gages are graduated in various ways. Some have an ordinary scale, as in Fig. 2, which requires the hight of both levels to be read and then the lower subtracted from the upper to obtain the difference in level. Others are graduated as shown by Fig. 3, the purpose being to avoid two readings. A U-tube supplied with this scale must be filled until the level on both sides is at the zero graduation. Either the upper or the lower scale is read and the reading is doubled to obtain the inches of water. If any of the water is evaporated or otherwise accidentally lost, however, the reading will be inaccurate.

Another form of scale is shown by Fig. 4, which obviates doubling the reading. This may be made adjustable through a small range so that its zero may be readily brought to coincide with the no-pressure position of the water level.

A gage giving direct readings may be made as shown in Fig. 5. This consists of a cup *C* of comparatively large diameter, containing liquid in which a glass tube of small bore is placed. As the liquid rises in the tube the level in the cup sinks, but on account of the large area of the liquid in the cup its level does not sink appreciably. Thus, if the cup is 2 inches in diameter and the tube ⅛-inch bore, for each inch the level in the tube rises that in the cup falls 1/256 inch. If the gage range is 5 inches, the total fall of level in the cup is about 1/50 inch, which in this case is negligible. If the cup is a closed chamber except for a tube leading into its air space, the instrument may be used as a differential gage.

It is on the principle of this gage that the mercury barometer is made, the form of the instrument being like Fig. 5 except that the glass tube is closed at the top and air excluded, as in Fig. 6. This is an absolute-pressure gage as distinguished from a differential gage, the absolute pressure of the atmosphere being recorded by the hight of the mercury column, as there is no other force in the tube than that due to the weight of mercury. A thumbscrew *D* is arranged to adjust the level in the open vessel at a fixed hight shown by the point *E*.

The simple U-tube may be used to show the difference of pressure in two parts of a closed pipe by connecting its two sides, one to each part, as shown by Fig. 7, instead of leaving one side open to the atmosphere. When measuring forced draft it should be connected in this way, one side to the ashpit, the other to the chimney flue. The pressure in the ashpit is greater than that of the atmosphere; that above the fuel bed may be less.

In some appliances for measuring the flow of steam the U-tube may be used, its form being somewhat different for practical purposes. When mercury is employed water of condensation settles on one side of the mercury column and influences the reading. This may be corrected as follows: Considering Fig. 8, the true difference of pressure is

$$H + \frac{H'}{13.6}$$

inches of mercury; that is, the water column in inches is divided by 13.6 to obtain the additional pressure in inches of mercury. If the water were on the other side it would be subtracted.

As applied to furnace drafts, the U-tube as described is not satisfactory because ordinarily, with natural draft, the difference does not exceed a few tenths of an inch. Therefore the instrument is difficult to read closely and does not indicate small changes of draft. To overcome this difficulty a great many ingenious devices have been constructed.

It would appear at first that the difficulty could be overcome by employing a liquid lighter than water. It is not practicable, however, to obtain one enough lighter to magnify the difference of level sufficiently, but two liquids, differing from each other in density, will answer the purpose. Thus in Fig. 9 assume the upper liquid to be oil whose specific gravity is 0.9 and the lower liquid water. If the levels in the cups *F F* are the same, then there is a difference of pressure corresponding to the difference in the weights of the column *H* of oil and water. Above *I I* the columns of oil, being of equal hight, balance each other; below *J J* the water columns also balance. Between *I I* and *J J* there is a column of water acting against the flue pressure and one of oil, equal in hight, acting with the flue pressure. As the pressure of the oil is 0.9 that of the same hight of water the draft is equal to $(H - 0.9\,H)$ or $0.1\,H$

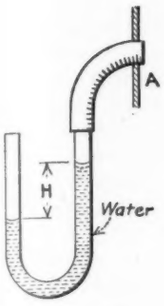


Fig. 1

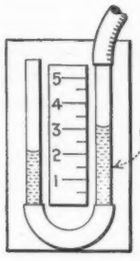


Fig. 2

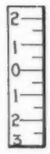


Fig 3

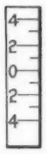


Fig 4

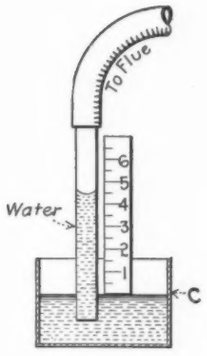


Fig. 5

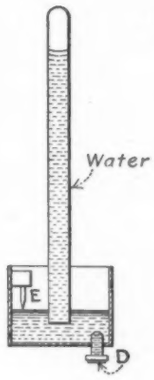


Fig. 6

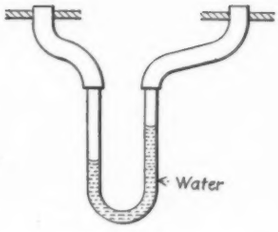


Fig. 7

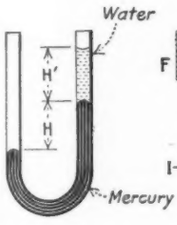


Fig. 8

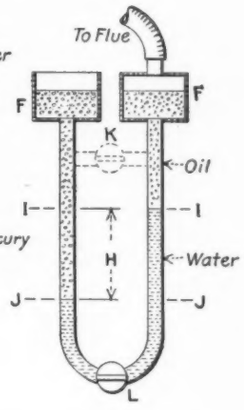


Fig. 9

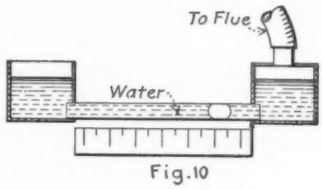


Fig. 10

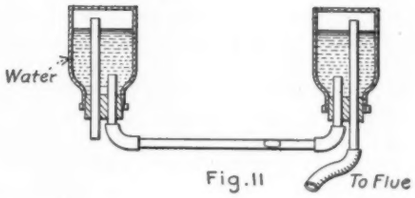


Fig. 11

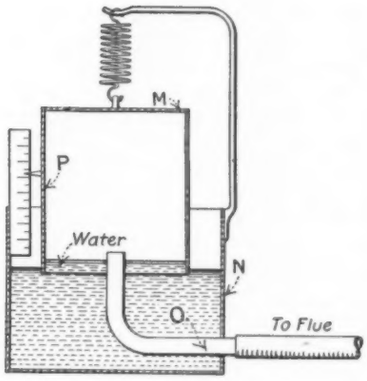


Fig. 12

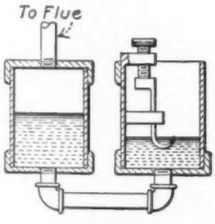


Fig. 13

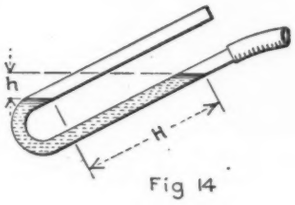


Fig 14

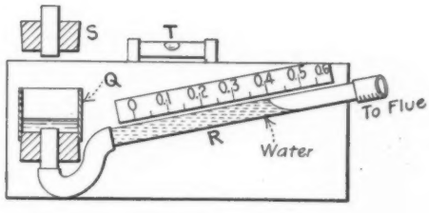


Fig 15

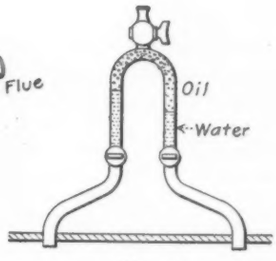


Fig 16

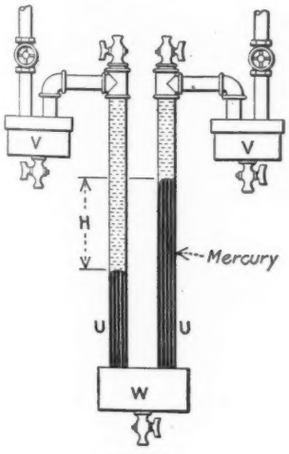


Fig. 17

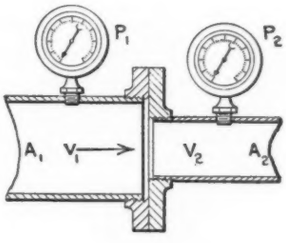


Fig 18

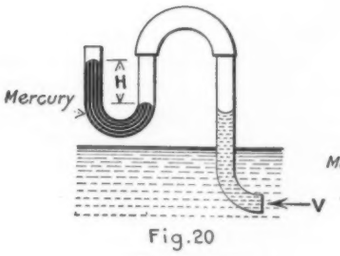


Fig.20

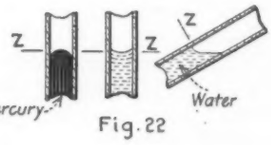


Fig.22

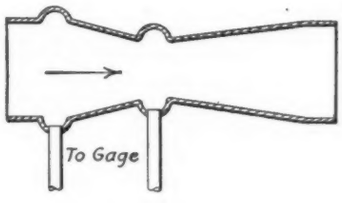


Fig 19

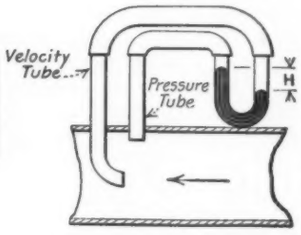


Fig.21

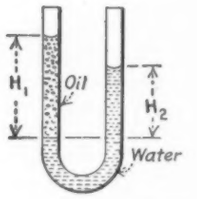


Fig.23

VARIOUS FORMS OF GAGES

inch of water; that is, the draft in inches of water is one-tenth of the reading. In general, calling the specific gravity of the oil S and h the draft in inches of water,

$$h = (1 - S)\ H$$

If the heavier liquid is not water and has a specific gravity S_1, then

$$h = (S_1 - S)\ H$$

By choosing the liquids for such a gage of nearly the same densities, any desired range may be obtained for the measurement of a small difference of pressure. The more nearly equal in weight they are the greater the motion of the liquid for a given draft. In general, the number of inches of rise in the ordinary U-tube is multiplied by $\frac{1}{S_1 - S}$.

In practice, when used as a draft gage, the difference of level in the cups $F\,F$ would alter the foregoing relation. To keep the cup levels the same a cross-connection with a cock K joins the tops of the tubes. Another cock L is placed at the bottom of the tubes. When in use K is closed first and L opened; when the oil stops rising in the right-hand tube, L is shut and K opened and the flue connection is removed. This allows the oil in the cups to come to the same level again without disturbing the levels in the tubes. The cocks are then reversed until the oil has risen again as high as it will, and the level in the cups is adjusted once more. This is repeated until the oil no longer rises, upon which the reading is taken. This is the principle of Hoadley's draft gage.

The draft gage shown in Fig. 10 consists of two cups, one closed, the other open to the atmosphere. They are connected below the water line by a horizontal tube of small bore containing water in which there is a small air bubble. The motion of this bubble is a measure of the draft. As the water rises in the right-hand cup the bubble moves to the right a distance proportional to the difference of level in the cups. By making the ratio of cross-sectional areas of the cup and the tube bore the required amount, any desired range of motion may be obtained, the motion being inversely proportional to the ratio of areas.

Although the writer has never used this form of draft gage, he believes it would be unreliable on account of the possibility of the bubble changing its position in the tube irrespective of the draft. As the bubble moves toward the right the water which wets the wall of the tube tends to remain stationary. This objection could be overcome by using mercury instead of water, in which case, however, it would be necessary to make the ratio of cross-sectional areas very much greater for a desired motion of the bubble.

Fig. 11 shows how this form of gage may be easily made from two bottles with perforated corks together with glass and rubber tubing.

A draft gage designed by Professor Kent is shown by Fig. 12. It is made of two tin vessels M and N, the former inverted and suspended within the other by a spring. Its interior is subjected to flue pressure through the tube O. The pressure of the atmosphere being greater tends to push down the suspended can. This tendency is opposed by the spring and, according to its extension recorded by the pointer P, the draft is measured. The equation of the instrument, neglecting buoyancy, is

$$p \times a = 0.036\ h \times a = n \times H$$

from which,

$$h = \frac{n \times H}{0.036 \times a} = k \times H$$

where,

p = Difference between flue and atmospheric pressures in pounds per square inch;

a = Internal cross-section, in square inches, of can M;

n = Number of pounds corresponding to 1 inch extension of spring;

H = Extension of spring in inches corresponding to $p \times a$ pounds;

h = Draft in inches of water;

$k = \frac{n}{0.036\,a}$.

The equation ($h = k\,H$) shows that to obtain the draft the reading on the scale is multiplied by a constant which is less than one. The increasing buoyancy as M sinks is negligible if the design of the gage is good. This instrument does not quickly respond to changes of draft, however, and is cumbersome.

Fig. 13 shows Miller's draft gage in which the difference of level is not magnified, but a very precise instrument, the hook gage attached to a micrometer, is used to measure the level.

Perhaps the simplest and most satisfactory form of draft gage is the variation of the ordinary U-tube shown by Fig. 14. It is set on a slant so that a vertical difference of h inches between the water levels causes an actual motion of H inches. By varying the slant any desired magnification may be obtained. This is generally made with a scale on the principle illustrated in Fig. 4 so that only one reading need be taken. It may be readily constructed of a piece of gage glass Q (Fig. 15) and a glass tube R together with some rubber tubing; perforated corks S may be used if forced draft is to be measured. The gage should be mounted on a board to which a spirit level T is attached so that the slant of R may be kept as intended.

For the purpose of making this gage more compact the slanting tube is sometimes in the form of a helix.

Most of the previously described gages are applicable to the measurement of water as well as gas pressure, the liquid used in them for this purpose being mercury or oil. Fig. 16, for example, shows the ordinary U-tube inverted for use with oil. The interpretation of the reading is the same as explained for the Hoadley gage. It is important to exclude air bubbles from the piping system of such a gage, and this can be done by carefully introducing the oil or mercury through cocks. A column of air in the connections on either side would cause a reduction of weight which would be balanced only by a variation of the true level difference of the gaging liquid.

Fig. 17 shows a gage using mercury for large differences of pressure. One-eighth-inch pipe and fittings are used and the long gage glasses $U\,U$ are joined by a cast-iron well W. The pots $V\,V$ are arranged to catch the mercury should the pressure become sufficiently great to blow it beyond the gage glasses. The cocks are arranged for introducing the mercury and excluding the air. In the right-hand column there is acting a weight of H inches of mercury which is partly balanced by the same hight of water on the other side. The mercury column causes a pressure of $13.6\,H$ inches of water; therefore the pressure difference indicated by the gage is

$$13.6\,H - H = 12.6\,H \textit{ inches of water}$$

or $1.05\,H$ feet. That is, to interpret the reading of this gage its reading in inches must be multiplied by 1.05 to get the difference of pressure head in feet.

Applications of the Differential Gage

In Fig. 18 the symbols P, V and A stand for pressure head in feet, velocity and area in square feet respectively at the two sections of the pipe. Since the sections at A_1 and A_2 are different, the velocities will be different, V_2 being as many times V_1 as A_1 is times A_2. Neglecting the losses due to friction, etc., the total energy of the water in the one section is equal to that in the other, this energy being in two forms, pressure and velocity. Since the velocity in the small pipe is greater than that in the large, the pressure in the latter must be less and the diminution in pressure energy must equal the gain in kinetic energy. Therefore, the difference of pressure in the two pipes is a measure of the velocity and consequently the quantity, according to the following relation:

$$Q = \frac{A_1 \times A_2}{\sqrt{A_1^2 - A_2^2}} \sqrt{2g\,(P_1 - P_2)}$$

in which Q is the quantity flowing in cubic feet per second and g is the acceleration due to gravity, 32.16. This is the principle of the venturi meter, a very precise instrument for the measurement of the flow of water. It has also been

applied to steam and perfect gases but with less success. Fig. 19 shows the usual proportions of the meter, the actual discharge being between 95 and 99 per cent. of that shown by the formula.

Fig. 20 represents a Pitot tube, another instrument for measuring velocities and quantities. It is a tube having a quarter bend, set facing the direction of flow. The hight H of the liquid in the tube is a measure of the velocity or

$$V = \sqrt{2gh}$$

h being the head in feet of whatever liquid or gas is flowing. To convert the reading H to the equivalent head of the fluid whose velocity is being measured, it is necessary to multiply by the ratio of the densities of the gaging and the measured fluids. Thus, if the gaging liquid were 10 times as heavy as the fluid being measured, h would equal 10 H. In general,

$$V = \sqrt{2gCH}$$

in which C is the ratio of densities.

The Pitot tube has been used largely for velocity measurements in open streams, but not until comparatively recent years has it been applied extensively to the flow of liquids and gases in closed pipes. Fig. 21 shows its arrangement for this purpose. Since in such a system pressure is recorded as well as velocity, a pressure tube is added. Static pressure only acts through this tube and balances the static pressure in the other tube; hence the gaging liquid records the velocity head only.

The simplicity of this device is alluring, but it should be used with caution in careful measurements as a great many possible conditions may cause inaccuracies. The velocity of the fluid varies across the pipe section and the velocity tube must be placed at the point of mean velocity. The fluid sweeping past the pressure tube tends to make a suction reducing the effective pressure in it. If there are eddies in the current these also tend to distort the record. The gaging generally should be made in a clear, straight stretch of pipe where there is nothing to disturb the uniform passage of the fluid.

When used for the flow of water with mercury as the gaging fluid

$$V = 8.23\sqrt{H}$$

and for the flow of air at atmospheric pressure and 60 degrees Fahrenheit, water being the gaging fluid,

$$V = 66.3\sqrt{H}$$

in which V is the velocity in feet per second and H the reading in inches.

The gages described are useful in measuring losses of pressure in various parts of blower systems. By their use leaks may often be located and otherwise unsuspected losses remedied.

Practical Hints

The level of mercury in a glass tube curves from the center downward, while that of water curves upward. It is best to read the hight on a horizontal line tangent to the curve, as $Z\,Z$, Fig. 22. In the inclined-tube form of gage, if the tube is about ⅛ inch in diameter, the level takes a curve as shown, which is very conveniently read at Z. The bore of the tube used should not be too small as capillary action may cause error.

When oil is used as a gaging fluid its specific gravity may be found as shown by Fig. 23. A little oil is floated on water in one side of a U-tube and its hight H_1 and that of the column of water balancing it H_2 are measured. Then the specific gravity of the oil is $\frac{H_2}{H_1}$.

Considering the units of pressure measurement dealt with, it is unfortunate and illogical that three different physical quantities and two starting points for measuring are employed when one would do. There is no reason why the ordinary spring-pressure gage should not be graduated in pounds above a perfect vacuum and its units used for whatever pressures are measured. U-tube scales, no matter what their liquid, could be graduated to read directly in pounds per square inch quite as conveniently as the present system of inches. It may be contended that for drafts and small pressure differences the pound is too large a unit. This could be remedied by using one one-hundredth of a pound as the unit for small pressures. There would then be 3.6 of these units to an inch of water.

Cost of Power in an Office Building Plant

By A. L. Sweetser

Many office buildings in our large cities have a daily population of 500 to 2000 people and in order to supply their requirements in the way of light, heat, air, water, transportation and toilet privileges a power plant is necessary.

Such a plant may be said to be the very heart and lungs of the building and upon its efficiency depends to a great extent the rental of the numerous offices.

Recently the author was requested to examine such a plant for some clients. The results of this investigation may be of interest.

The building was a 16-story structure, of brick and steel, containing 100,000 square feet of rentable floor space.

The boiler plant consisted of three horizontal water-tube boilers rated at 150 horsepower each; the furnaces were equipped with mechanical stokers, and the average horsepower developed by the boilers was 196, on an evaporation of 7.09 pounds of water per ton of coal.

Three engines of the simple, high-speed automatic type were each connected to 100-kilowatt direct-current generators supplying electrical energy for the elevator service, lights and motor fans in the various offices throughout the summer months.

During the day the average electrical horsepower shown at the switchboard was 89, the indicated horsepower was 110, and the efficiency, therefore, 80.9 per cent.

The remaining equipment consisted of three feed pumps for the boilers, two service pumps to supply water to the upper part of the building and one 5-horsepower ammonia ice machine for cooling the drinking water. A vacuum system using the exhaust steam from these units heated the building.

There were three Otis electric elevators which made an average of 1000 trips per day and carried approximately 2790 passengers. The power required to operate these elevators was 17.6 per cent. of the total amount generated. From records it was shown that the average steam consumption of the generator engines amounted to 107 horsepower. The remaining horsepower was divided among the feed and service pumps, the vacuum system, the refrigerating plant, etc., as given in the following table:

DISTRIBUTION OF POWER

Average horsepower generated by boilers	196	
Average horsepower consumed by engines	107	
Average horsepower consumed by vacuum pumps	16	
Average horsepower consumed by feed pumps	23	
Average horsepower consumed by service pumps	6	
Average horsepower consumed by refrigerating plant	5	
Average horsepower consumed by elevators		17.15
Average horsepower consumed by lights and fans		21.85
Total	196	

COST OF ELEVATOR SERVICE PER MONTH

Cost of power	$95.25
One starter	60.00
Three operators	150.00
Maintenance	5.00
Total	$310.25
Cost per day	10.37
Cost per trip	0.01037

The following tables give the monthly cost of power and the total cost for the building:

MONTHLY COST OF POWER

Cost of coal	$476
Chief engineer	120
Two assistants	100
Two firemen	100
One oiler	45
Repairs	56
Total	$897
Cost per boiler horsepower per month	$4.57
Cost per boiler horsepower per day	0.152

TOTAL COST OF POWER FOR THE BUILDING

Cost per month	$897.00
Cost per day	26.30
Cost per year	9,599.50
Rentable floor space, 100,000 square feet.	
Cost of power per square foot of space per year	0.95
Yearly cost of elevator service	4,226.70
Total cost with elevator service	13,826.20
Total cost per square foot of space per year	0.138

Inertia of Air Compressor Intake

By Snowden B. Redfield

An attempt to show, by applying the theory of inertia forces of reciprocating parts of an engine, that the distinct increase in air pressure at the end of intake stroke, often observed on air-compressor indicator diagrams, is due to inertia of the moving air column. Possible application to practical purpose of materially increasing volumetric efficiency of compressor.

That the inertia of the pulsations of air entering a compressor cylinder, due to the reciprocating action of the piston, may cause an increase in the pressure of the air at the end of the suction stroke, appreciably above atmosphere, thereby increasing the volumetric efficiency of the cylinder, is a subject of considerable discussion among pneumatic engineers. Fig. 1 shows an air-compressor indicator diagram with this rise in intake pressure above the atmospheric line at the end of the suction strokes, *A A;* such diagrams often being found and by some mistakenly regarded as evidence of discharge-valve leakage.

It is quite generally admitted that, under favorable conditions of unobstructed piping, high piston speed and relatively large number of reciprocations, a water pump actually will deliver more water than its piston displacement would indicate. This is simply because the water attaining a high intake velocity, does not stop instantly when the piston reaches the end of its travel, but the inertia of the moving water tends to continue the flow during the instant of rest previous to reversal of piston stroke.

This sounds like "perpetual motion," but an instant's reflection will free the mind of such an impression. The speed and inertia given to the rapidly moving water must, of course, have come from the piston and this latter must have received them from the power of the driving mechanism, so that every extra foot-gallon of work done will be accounted for at the motor end of the machine.

Such conditions in a water pump could obtain only with relatively low heads because, water being incompressible, the energy of flow quickly would be absorbed in overcoming the pumping head as the water rushed directly through the pump cylinder into the discharge. This restriction would not hold, however, with moving air, for, although the air is very much lighter and has much less inertia for a given volume, even at its higher speed, this inrushing air would not encounter the pressure of the discharge at all; but simply would crowd into the cylinder against a pressure approximately atmospheric, causing a slight increase of this pressure above the atmospheric line, as shown in Fig. 1.

If the inertia effect does take place, as the rise in the air-intake line would seem to show, to marked degree sometimes, what effect does it have, and can any practical steps be taken to improve the action of a compressor thereby? The effect of an increase in the pressure of the air just as the compression stroke commences, is materially to increase the volumetric efficiency of the cylinder; that is, to increase the amount of air compressed by a given sized cylinder. In these days of obtaining the finest points of economy, "every little bit helps" and if, by an inexpensive arrangement of piping, an increase of a few per cent. in volumetric efficiency may be obtained, this may be counted a material gain.

Basis of Calculation

This article is not intended to be a complete theoretical explanation of this phenomenon, but an attempt to point out in a more or less crude fashion that there is a theoretical reason for accounting for its occurrence on the inertia basis. Probably some mathematical expert can figure it out much more directly and accurately.

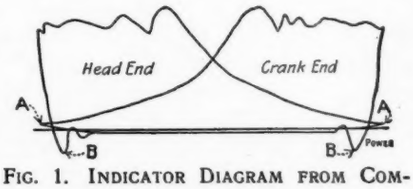


Fig. 1. Indicator Diagram from Compressor, Showing Effect of Air Inertia

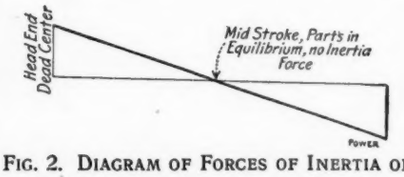


Fig. 2. Diagram of Forces of Inertia of Reciprocating Parts with Scotch Yoke

As a basis for calculating such an effect of inertia we may turn to the method of calculating the inertia forces of the reciprocating parts of a steam engine. We may do this on the assumption that the motion of the air in the intake pipe is in a series of pulsations corresponding to the reciprocating motion of the air piston. Close to the compressor cylinder, this probably is very nearly true and the effect must be similar in a gradually decreasing degree at greater distances. Even at the further end of a compressor intake pipe of considerable length, these pulsations are distinctly noticeable by placing the hand in the current of air rushing into the pipe. Far enough away from the piston, the pulsation waves are, no doubt, distinctly modified from the approximately simple-harmonic motion of the piston, but the distance would have to be very great before the elasticity of the incoming air would be such as to absorb all pulsations and result in an even flow of air into the pipe.

Those who have studied the crank-effort diagram of the steam engine, or other machine involving a crank and connecting rod, are aware that the starting and stopping forces at the ends of the strokes are, with an infinite connecting rod (or a "Scotch yoke"), equal to the centrifugal force that would be produced if all the reciprocating parts were revolving about the shaft at the radius of the crank circle. If the force at the end of the back stroke or beginning of the forward stroke, which tends to produce tension in the rods, be considered positive, the stopping force at the other end of the stroke, tending to produce compression in the rods, must be considered negative.

At some time, then, during the stroke of the piston, the inertia forces must become zero, the parts having been completely accelerated and traveling along in equilibrium. With an infinite connecting rod (Scotch yoke), this point of zero inertia force occurs at mid-stroke, and a diagram of the forces of inertia of the parts would be as in Fig. 2. In this case, the forces at each dead center would be alike, and the point of zero force would be, as said, at mid-stroke.

In this reasoning the difference between *inertia* and *inertia force* must be distinctly appreciated. At mid-stroke the *inertia,* or stored energy, would be maximum, due to the high velocity; but the *force* exerted would be zero, because there would be no acceleration or retardation. Toward the stroke ends, however, the slowing of the motion would transform the *inertia* into an active *force* pressing on the crank pin and assisting the motion, this force becoming a maximum just at the instant of stopping. At this instant, of course, the *inertia* would be zero. Whatever force assists the crank pin toward the end of each stroke, correspondingly tends to retard the crank pin during the start of the next stroke, so no net work is done by these forces.

Effect of Connecting-rod Angularity

For the real conditions of a finite connecting rod, we may refer to some calculations by Professor Jacobus in the *Transactions* of the American Society of Mechanical Engineers, Volume 11, pages 492 and 1134. In these papers will be found a table of factors worked out, by which the theoretical, infinite-rod forces may be multiplied to obtain the actual, finite-rod forces. A plot of such a calculation will produce a curve like Fig. 3, where the forces at the two ends of the stroke are unlike and where the zero point is somewhere around 80 degrees of crank angle, measured from the head center. These changes are due simply to the effect of the connecting-rod angularity in putting the piston forward of where it would be at any one time if the rod were infinite and had no angularity.

For an infinite rod, the force of inertia, in pounds, at any position of crank angle is expressed as follows:

$$F = \frac{W \pi^2 N^2 R}{900\, g} \cos.\ \theta$$

where

W = Weight of reciprocating parts, in pounds;
N = Number of revolutions of crank per minute;
R = Crank radius, in feet;
g = Acceleration of gravity, 32.2;
θ = Crank angle, measured from head dead center.

When $\theta = 0$, $\cos.\ \theta = 1$ and F becomes the same as the centrifugal force, as already explained, for the dead-center position.

From Professor Jacobus' figures, for a machine having a connecting rod of length equal to five times the crank arm, the usual design for air compressors, the force at beginning or end of either stroke, should be

$$F = \frac{W \pi^2 N^2 R}{900\, g} (\cos.\ \theta + 0.2)$$

Application to Moving Column of Air

To apply this formula to a moving column of air in a pipe, W will be the weight of air in motion, N will be the number of double reciprocations corresponding to the revolutions of the compressor crank, but the value of R will not be the crank radius. This will be understood from the fact that, due to the intake pipe being much smaller in area than the air cylinder, the air in this pipe must travel faster and further at each stroke than the piston does. Suppose the intake-pipe area is approximately 12 per cent., or ⅛ of the piston area. Then, at any given time, the air must be traveling 8 times as fast as the piston and, in order to fill the cylinder, the requisite air will have to travel 8 times as far in the pipe as the piston does in the cylinder. In other words, the air pulsations are 8 times as long as the piston stroke, and consequently in this case the value of R in the formula for the air must be 8 times the length of the compressor crank arm.

As might be expected, the formula shows that the inertia force is proportional to the square of the number of revolutions. This means that the greater the number of stops and starts in a given time, the greater the forces. Consequently a relatively short stroke with a given piston speed is conducive to heavy forces.

An Example

To take a specific example: Suppose we have a 36-inch stroke air compressor, running at a speed of 100 revolutions per minute, a practical figure for modern, high-speed practice, especially for direct-connected electric drive with motor on compressor shaft. Let the cylinder be 34 inches in diameter, and let its average net area be 900 square inches. The intake-pipe area, being ⅛ of this, let its inside diameter be 12 inches, with an actual area of 113 square inches. The ratio of cylinder and pipe areas then will be 8 to 1, and so the air speed and length of air pulsation will be 8 times as great as those of the piston.

Let us say that the intake pipe is 25 feet long, from entrance at outside of building to cylinder; let the temperature of the air be 60 degrees Fahrenheit. At the instant of stoppage, the air in the inner end of the pipe will be compressed by the inertia to some pressure above the atmosphere, while that at the outer end will be atmospheric and only that in the middle section will be, say, half a pound below atmosphere. It then will be reasonable to assume that the average pressure of all the air is atmospheric at the instant of stoppage and greatest force and pressure. The weight of the air contained in the pipe then must be 0.0764 pound per cubic foot, and as the volume of the pipe of 12 inches inside diameter and 25 feet length is 19.6 cubic feet, the total weight of air flowing in the pipe at this instant will be 1.50 pounds.

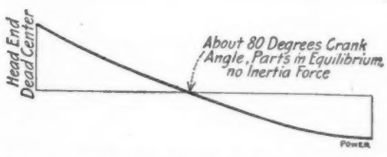


Fig. 3. Similar Diagram with Connecting-rod Length Five Times the Crank Arm

Taking, first, the force at the head center, we have, by applying the formula already given for a five-crank length connecting rod:

$$F = \frac{1.50\, \pi^2\, 100^2 \times 1.5 \times 8 \times 1.2}{900\, g}$$

In this expression, the factor 1.5 is, of course, the crank radius in feet, and the factor 8 is the ratio of cylinder area to intake-pipe area and, consequently, the ratio of air-pulsation length to piston stroke; so that the product of these two factors represents the imaginary length of crank arm that would produce the air pulsations occurring in the intake pipe. The factor 1.2 is the sum of the cosine of 0 degrees (unity), the crank angle at head dead center, and the factor 0.2, calculated by Professor Jacobus for the connecting-rod length chosen.

The solution of the foregoing expression is:

$$F = 73.5 \textit{ pounds}$$

Inertia Forces Due to Reciprocation

This is the total pressure exerted in stopping the column of air at the end of the suction stroke at the head dead center, but it must be remembered that, if the air passage is continuously of the same area, 113 square inches, this pressure will be distributed over this whole area and so the pressure per square inch exerted will be $\frac{73.5}{113} = 0.650$, or over ⅝ of a pound.

Cumulative Effect

Just at this point we should consider a further inertia effect: the "piling up" and compressing of the air in the cylinder and the inner portion of the intake pipe. The result of such an action would be to allow the remainder of the air in the pipe to continue flowing and crowding in, so that the actual length of air pulsation is greater than we have assumed. The effect of increasing this pulsation length is to increase the value of R in the formula, and thus further increase the force which we are endeavoring to calculate. The logic is that, the number of pulsations per minute remaining the same, the stopping force must be greater if the distance traveled, and consequently the linear speed, is greater; also the additional air entering the pipe adds its inertia to the effect.

As this would tend further to increase the pressure, let us assume that the final pressure would be, say, 0.7 pound above atmosphere, or 15.4 pounds absolute. Now, the volume of the compressing cylinder is 18.7 cubic feet and, as an estimate, we may include the last 6 feet of intake pipe in this "piling up" effect. This gives a total volume of 23.4 cubic feet into which the incoming air is crowded from an initial pressure of 14.7 to 15.4 pounds. Then this 23.4 cubic feet of air will occupy a space of

$$23.4 \times \frac{14.7}{15.4} = 22.3 \textit{ cubic feet}$$

thus leaving

$$23.4 - 22.3 = 1.1 \textit{ cubic feet}$$

of the last part of the pipe for more air to crowd into. With a 12-inch intake pipe this means a length of 1.4 feet, and this length is to be added to the value of R, or the imaginary radius of the pulsation wave. As the old value of R was $8 \times 1\frac{1}{2} = 12$, we now have 13.4 feet for the more probable value of R.

As the value of F is directly proportional to R, the more probable value of F now will be

$$0.650 \times \frac{13.4}{12} = 0.726 \text{ pound}$$

giving a final intake air pressure of over 15.4 pounds per square inch absolute, which agrees with our assumption when we began to consider this "piling up" effect. It is, therefore, reasonable to assume that there would be, under the circumstances of this case, an initial air pressure just at commencement of compression, of 0.726, or nearly ¾ of a pound above atmosphere. This would increase the volumetric efficiency by quite 5 per cent., a result well worth striving for.

Forces Unequal at Head and Crank

At the crank end of the stroke, the force would be somewhat less, as the value of *cos.* θ would be — 1, and — 1 + 0.2 = — 0.8. This would then make the pressure at the crank end about 0.484, or a little less than ½ pound per square inch above atmosphere, increasing the volumetric efficiency at this end by over 3 per cent. The average efficiency increase for the two ends then would range about 4 per cent.

This condition of unequal effects at the two ends is borne out in practice, for an examination of the indicator diagrams containing these inertia effects invariably shows more initial pressure at one end than at the other.

To be strictly logical, the increased quantity of air admitted to the outer end of the pipe by the crowding and compressing of that at the inner end, should be taken into account. This increase was shown to be 1.1 cubic feet, bringing the weight up to 1.58 pounds. This would further increase the head-end inertia force to

$$\frac{1.58}{1.5} \times 0.726 = 0.765 \text{ pound}$$

per square inch, or over ¾ of a pound pressure; and the crank-end force to 0.51, or over ½ pound per square inch. This would tend to show that we are at least on the conservative side, leaving room for pipe friction and other losses.

Practical Considerations

From the foregoing, it would appear that a longer intake pipe would contain more air in motion, and so would give an increased inertia force and higher volumetric efficiency. Double the length of pipe would give double the weight of air; but, as before intimated, the pulsations probably are modified considerably at the end of so long a pipe. Another matter too, is the loss of pressure by friction through this long pipe, but this is really negligible if the air speed is kept down around 4800 feet per minute. As an example, tables worked out in Kent, from B. F. Sturtevant Company's formulas, show that the loss of pressure through 25 feet of 12-inch pipe, with a speed of 4800 feet per minute, is about ½ ounce.

It may well be asked "why should the pipe produce this inertia effect any more than if the air flowed directly from the atmosphere into the cylinder, thus avoiding even the small friction of flow. The answer is that the air in the pipe has a smooth flow and has an opportunity to attain the velocity calculated; whereas, if the pipe were absent, the atmospheric air would flow from all sides at low speed and, furthermore, all energy of motion would be lost in eddying at entrance. The pipe keeps the air flowing straight, swiftly and without eddying to any great extent.

Some persons will realize that this force or pressure required to stop the incoming air at the end of the stroke must be balanced by an equal and opposite inertia force required to *start* the pulsation at the beginning of the stroke. No better evidence of this can be desired than the ever present "hooks" *B B*, Fig. 1, on all air-compressor indicator diagrams. Two excuses usually are given for these: inertia of indicator parts dropping from the pressure of discharge to that of intake, and the pressure required to open poppet valves. The first excuse seems inadequate with a modern, light indicator and as to valve resistance, the writer has seen these "hooks" with Corliss inlet valves open wide before the stroke started.

It will be well to add that an actual case of this kind; but which, owing to several elbows and a strainer in the intake pipe, it was impossible to figure upon intelligently, recently has been brought to the writer's attention, where the initial pressure of the intake line is about 1½ pounds above atmosphere at one end and 1 pound at the other.

Pumps and Pumping Calculations

By C. F. Swingle

General remarks concerning the classification and operation of pumps, together with a few simple formulas for calculating the dimensions, capacities, head, etc.

In the ordinary single-cylinder and duplex steam pumps the steam is admitted during the full stroke of the piston because of the constant resistance of the water. This means that the terminal pressure equals the initial pressure; consequently, the steam consumption per horsepower greatly exceeds that of an engine in which advantage can be taken of the expansive properties of the steam.

There are, however, various methods of still further utilizing the heat energy of the steam consumed by a steam pump; one of which is the addition of a second cylinder of larger diameter into which is led the exhaust from cylinder No. 1. If the areas of the two pistons are properly proportioned, the steam exhausted from cylinder No. 1 (high pressure) into cylinder No. 2 (low pressure) may be made to do the same amount of work in the latter that it did in the former. This is the principle of the compound steam pump, and the saving thereby effected ranges from 20 to 30 per cent., depending upon whether the steam in the low-pressure cylinder exhausts into the atmosphere or into a condenser. The usual number of expansions is from two to four, depending, also, upon whether the low-pressure cylinder is run condensing or noncondensing. But the same economy in the use of steam cannot be expected of compound direct-acting steam pumps as that realized from the compound engine, the latter having the advantage of a heavy flywheel and the momentum of the moving parts.

The duty of a pump is sometimes figured as the number of foot-pounds of work done for each 100 pounds of coal burned per hour; this is found as follows:

Duty = pounds of water raised per hour $\times$ *total lift in feet* $\times$ 100 $\div$ *pounds of coal burned per hour*

In this expression the total lift means the vertical distance in feet from the surface of the water into which the suction pipe enters, to the surface of the water in the tank or reservoir into which the water is discharged. In determining the suction lift the distance from the surface of the water to the suction chamber may be measured, or if there is a vacuum gage attached to the suction chamber, each inch of vacuum equals approximately one-half pound of atmospheric pressure removed. Suppose, for instance, that the vacuum gage shows 20 inches; then the suction pressure will be 10 pounds, and as each pound of pressure represents a head of 2.3 feet, the suction lift will be:

$$\frac{20 \times 2.3}{2} = 23 \text{ feet}$$

Having noted the water pressure as indicated by the pressure gage (assuming this to be 100 pounds), the discharge head equals

$$100 \times 2.3 = 230 \text{ feet,}$$

and total lift equals

$$23 + 230 = 253 \text{ feet.}$$

The horsepower required to raise water to a given hight may be found by the following rule:

Horsepower = number of gallons per minute × hight, in feet, ÷ 3957

The number of pounds of steam used per minute by a simple, direct-acting pump equals

(Diameter of steam cylinder, in inches)² × stroke, in inches, × density of the steam × number of strokes per minute ÷ 2200

For duplex pumps use the number of strokes made by both pistons. The density of the steam refers to its weight per cubic foot at the pressure in the steam chest or cylinder. For a compound pump, use only the dimensions of the high-pressure cylinder. The gallons of water delivered per pound of steam used equals

0.0034 × (diameter of water cylinder, in inches)² ÷ [(diameter of steam cylinder, in inches)² × weight of one cubic foot of steam at the pressure used ÷ 2200]

The density of steam at various pressures may be ascertained by reference to steam tables.

To find the number of pounds of water delivered per pound of steam used, substitute 0.02835 in place of 0.0034 in the foregoing expression.

The hight, in feet, to which water can be raised with a given horsepower may be expressed as follows:

Hight = 3957 × horsepower ÷ number of gallons

The area of the water piston (or of both water pistons if a duplex pump is used) required to raise a given volume of water per minute, is

Area in square inches = gallons × 231 ÷ (length of stroke in inches × strokes per minute)

The length of stroke required to raise a given volume of water with a given number of strokes per minute may be expressed as,

Stroke in inches = gallons × 231 ÷ (area of piston × strokes per minute).

The piston speed, in feet per minute, equals length of stroke in inches × strokes per minute ÷ 12.

To find the area of the suction pipe required for any pump, use the following equation:

Area of suction pipe, in square inches, = area of water cylinder, in square inches, × piston speed in feet per minute ÷ 200

For the discharge pipe substitute 380 in place of 200.

The function of an air chamber on a pump is to insure a steady and uniform discharge, whereas the object of a vacuum chamber is to facilitate the changing of continuous into intermittent flow.

The hight in feet to which a pump can force water when taking water under pressure is

$$\text{Hight} = \frac{\left(\begin{array}{c}\text{2.3 area of steam piston, in} \\ \text{square inches, } \times \text{ steam pressure}\end{array}\right)}{\text{area of water piston}} + \text{pressure in suction pipe}$$

To find the pressure divide the hight by 2.3.

Developments in British Steam Plants

By James A. Seager

The salient features of a few devices which have recently been developed in British steam-engineering practice for the purpose of increasing the efficiency of the boiler plant. They include smoke consumers, a device for increasing the circulation of water in a boiler and a system of returning the water of condensation to a boiler.

The production of power by means of steam is still making rapid strides in Great Britain in spite of the increasing popularity of power and suction gas and the introduction of oil engines as prime movers. One of the reasons for this is the abundant supply of cheap coal to be found in Great Britain, which, with the recent improvements in equipment, renders it possible to utilize the poorer qualities of coal, such as slack and duff, which heretofore have had comparatively little commercial value. Another impetus which has recently been given to power-plant operation in Great Britain is due to the fact that engineers and manufacturers have, to a very large extent, abandoned the old and expensive methods of power transmission by means of belting and shafting and have utilized the advantages of the centralized power plant combined with the economical transmission of power to the point of application by means of electricity.

Undoubtedly one of the greatest factors affecting economy lies in the efficient utilization of the fuel in the furnace of the boiler. This movement has received additional impetus in Great Britain owing to the very stringent laws and heavy penalties which are enforced by most municipalities upon manufacturers whose chimneys emit heavy black smoke, and every effort is now being made to obtain perfect combustion by providing the proper supply of oxygen to the fuel.

An interesting smoke-consuming apparatus perfected by the Clayburn Engineering Company, of Manchester, is illustrated in Fig. 1 and, although here fitted to a Lancashire boiler, is applicable to any type of land or marine boiler. The apparatus consists of a group of steel tubes or coils placed in the flue passages, combustion chamber or uptakes, according to the type of boiler. These coils are employed for heating the air to a high temperature before reaching the fuel, which it does through a specially constructed distributing chamber located as shown in the illustration. The air is further heated in the distributing chamber, from which it is discharged in jets, intermixing thoroughly with the volatile gases and thus completing the process of combustion. On entering the furnace the air is at a temperature considerably higher than the water in the boiler and is, therefore, itself a steam raiser. Atmospheric air is forced or drawn into the pipe *X* through the coils to pipes *A*, *B* and *C*, the two latter being provided with valves to regulate the supply to the superheating chambers *D* inside the furnace and *E* at the bridgewall. Pipe *F* is connected with chamber *E* at the bottom, and at the top with the air box *G* underneath the brick arch. This air box is perforated with ⅜-inch holes opposite oblong openings in the front of the bridgewall, as is shown in the sectional view. Passing through pipe *C* and terminating in pipe *F* is a ¼-inch pipe supplying steam jet *J*, the object of which is to draw air through the louvers *H* and *I* into the chamber *E*. The amount of air is controlled by dampers *R* and *L*, thus furnishing an ample supply of oxygen which is evenly distributed and mixed with the hot gases. An alternative arrangement of the coil *X* is to carry it to the back of the combustion chamber instead of connecting at *A* and then through the combustion chamber to pipe *F*. Moreover, in some cases the coils are dispensed with and an injector connected at *Y*. On tests this apparatus has shown a considerable saving in fuel as contrasted with a set of steam jets over the fire, and in one particular test it showed a saving of 10.8 per cent. in fuel over the ordinary natural-draft arrangement.

Another device which is finding favor is the torpedo smoke preventer and heat distributer manufactured by the Yorkshire Boiler Company. This consists, as shown in Fig. 2, of a hollow torpedo-shaped structure composed of special firebrick blocks supported on firebrick brackets from the bottom of the flue. This structure is placed a short distance behind the bridgewall and in the direct path of the gases. Air is admitted to the center of the structure by means of a pipe extending through the ashpit and bridgewall, and this pipe is fitted with a regulating grid placed at the front of the boiler between the two ashpits. Under the action of the hot gases the firebrick rapidly becomes incandescent, and the air, which has been already heated in the interior of the torpedo, escapes through small openings at the top and unites with the unconsumed volatile matter in the gases. The economy of fuel from the use of this appliance is considerable as it converts the convected heat, which would otherwise pass away with the furnace gases, into a store of radiant heat in close proximity to the top flue plates and thus causes rapid steam evolution. From comparative tests which were made on a Lancashire boiler with and without the torpedo smoke preventer it was found that a saving of nearly 10 per cent. in the fuel consumption was effected by installing this appliance.

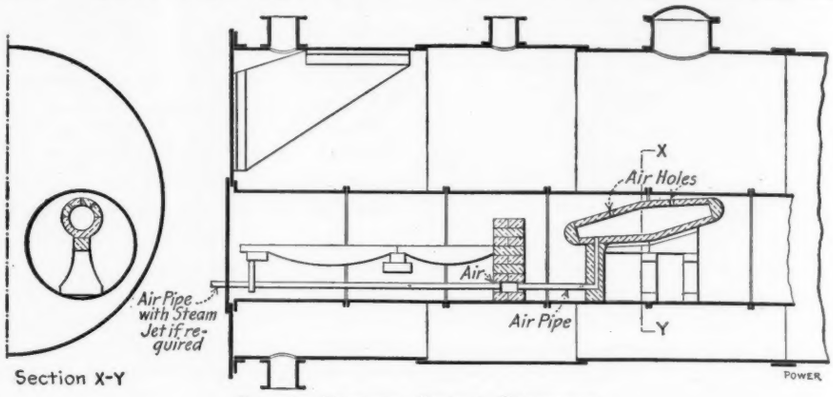
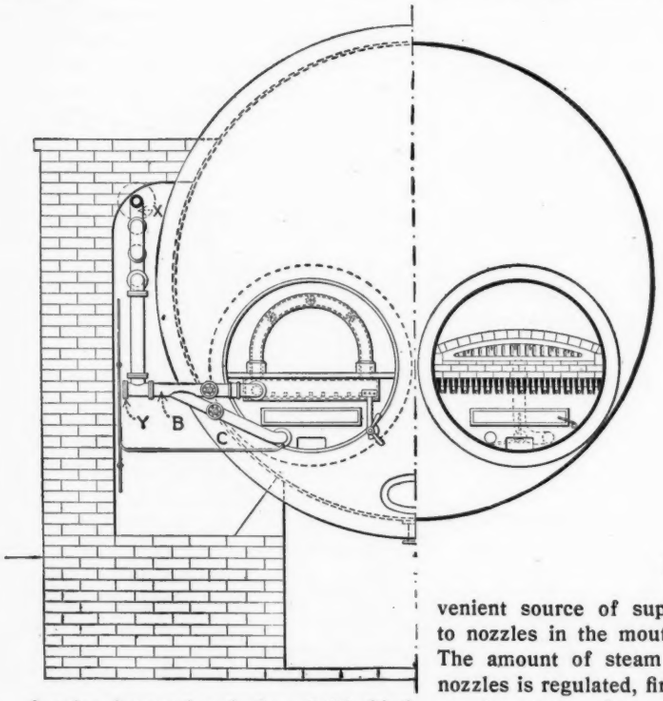


FIG. 2. TORPEDO SMOKE CONSUMER

Another interesting device of this kind is that manufactured under the Wilson and Furneaux patents by the Tyne Forced Draught Company which is illustrated in Fig. 3. It consists of a number of fire bars of special design placed across the furnace flue. These rest on three or more half tubes or troughs running lengthwise with the boiler, the upper edges fitting into grooves in the bottom of each fire bar. The bottoms of these half tubes rest in stands spaced about two feet apart in the furnace. There are three or more openings to the fire in each bar, and every bar is designed so as to form a separate air passage to the fire directly over each half tube which therefore acts as an air reservoir or duct for one-third of each bar in the furnace. A bell-shaped funnel is fitted at the front bar which is a different shape from the ordinary bars, as is also the back bar, the semicircular openings being omitted. At the front end of the half tubes or troughs a flap valve is fitted, which shuts off admission to them. The steam pipe from the boiler steam space or other convenient source of supply admits steam to nozzles in the mouth of each funnel. The amount of steam emitted by these nozzles is regulated, first, by a stop valve on the steam pipe, and, second, by a valve on the steam pipe connected with the flap valve on each half tube. Each half tube leads to a hollow in the bridgewall which is common to all, and on opening the flap valves at the front of these tubes a portion of the air induced by the steam jets is admitted to the hollow part of the bridgewall and from thence, after being heated, to the back of the wall where it mixes with the unconsumed products of combustion. The greater part of the air induced by the steam jets passes through the upper half of the passage formed by the half tubes and the lower portion of the fire bars and passes to the fire through the openings in each bar.

With this arrangement wing firing can be accomplished efficiently by a comparatively inexperienced fireman and the fires may be cleaned and every operation carried out without turning off the steam jets. Also, cheap small coal, duff and coke breeze can be burned effectively and the semi-automatic regulation of the air supply to the back of the bridgewall and to the furnace permits a maximum furnace efficiency to be attained with a minimum of smoke.



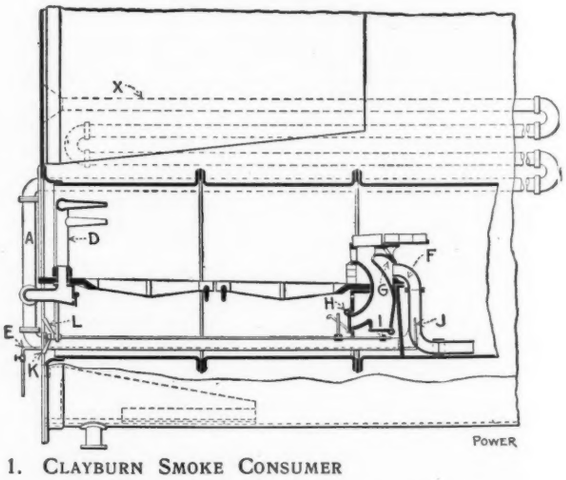


FIG. 1. CLAYBURN SMOKE CONSUMER

In addition to the problems concerning the thorough mixture of the gases and air in the furnace, considerable attention has recently been directed to the attainment of a thorough circulation of water in the boiler, it being claimed that this tends to reduce mechanical strains, especially in those types of boilers which have large areas of continuous surface exposed to local heat action. Also a good circulation secures more efficient and regular detachment of the steam from the water which is being evaporated. To

this fact is attributed the great popularity of the water-tube boiler in British land practice. Within certain limits, the faster the water circulates in a boiler the greater the emission of steam because the cooler portion of the water is periodically brought over the hottest part of the boiler. One of the most interesting developments in this line is the Leeds circulator, which is illustrated in Fig. 4, and which has for its object the continuous circulation of the water in a Lancashire or similar type of boiler by means of baffles suitably placed in the boiler. In the drawing the line *A* represents a sectional division plate, the upper portion of which stands 3 inches above the ordinary water level of the boiler, while the lower edge passes half way down the furnace flues. In conjunction with this, line *B* represents horizontal plates four feet long, one of which is carried between the flues and another on each side of the boiler between the furnace flues and the shell. Further to the rear is another vertical division plate *D*, the upper edge of which stands 10 inches or more above the working level while the lower edge is carried just below the flues. The feed pipe is carried the whole length of the boiler, but its only outlet is near partition *D* at the rear end of the boiler.

As active ebullition is continually taking place in the section of the boiler immediately over the furnace, a large part of the water impinges against and dashes over the plate *A*, assisted in so doing by the curved form of the plate. This plate prevents the water returning again and as the heat at this part of the boiler is not of sufficient intensity to evaporate all the water that passes over *A* the surplus descends by gravity, but the plate *B* compels it to move further to the rear and thus increases the length over which circulation takes place. As all the feed water enters beyond *D* the water falls to the bottom of the boiler and, assisted by the aspirative action of the evaporation taking place over the boilers, moves forward under the flues as indicated by the arrows and mixes with the circulating currents.

When the water reaches the furnace flues it is nearly the temperature of evaporation. This rapid circulation tends to keep the whole of the boiler at approximately the same temperature and thereby eliminates local distortion. It also has a marked effect upon the deposition of sediment.

A recent development invented by George Wilkinson has for its object the return to the boiler of the heat units which would otherwise be lost in the condensed steam. It is well known that such losses in steam pipes are very heavy and with the exception of one or two special methods of returning the water of condensation to the boilers the ordinary method of clearing the pipes of water is by means of steam traps which are more or less wasteful. The Wilkinson thermal column* is designed to eliminate these troubles and to restore the water of condensation in a heated condition to the boiler. It consists of two chambers, the lower one a sump and the upper one a receiver. The water of condensation from the steam range, separators, steam jackets, radiators, etc., drains by gravity into the sump, which is connected by a riser to the receiver, the latter being placed five or six feet above the tops of the boilers so that water of condensation can quickly gravitate back to them through a connecting pipe and nonreturn valve.

The principle upon which this device works depends upon the fact that as soon as a body of steam is cut off from the source of supply it cools rapidly and the pressure drops. When there is no water of condensation in the sump the steam has free access up the riser and into the receiver, but when sufficient water of condensation has drained into the sump the bottom of the riser is immersed; this, together with the receiver, is cut off from the source of steam supply and the steam already contained quickly drops in pressure; hence the water rises and flows into the receiver. As soon as the sump has been drained, the steam follows up the riser and establishes equilibrium of pressure on the water in the receiver. When this occurs the water is free to gravitate into the boiler through the check valve. The receiver is of such capacity that notwithstanding the period during which it is discharging to the boiler, the maximum amount of condensation formed in the steam range during each cycle is incapable of filling the receiver. Should the receiver and riser become full of water an automatic relief valve on the receiver allows the access of water to escape until the riser again becomes charged with steam, when the apparatus again resumes its normal work. Such flooding does not, of course, occur under ordinary conditions. The apparatus is so sensitive that the heat radiation through good nonconducting covering material is sufficient to insure its effective working. The condensed steam can be lifted from the sump and returned to the boilers with a maximum drop in temperature of less than one degree Fahrenheit per foot of lift. There is no outlet whatever to the atmosphere and the water is at a temperature close to that of the steam. Another important feature of the thermal column is that it works by differences in temperature and not by volume so that its efficiency is independent of the amount of condensation which may be taking place in the steam pipes, cylinders and other parts of the steam system.

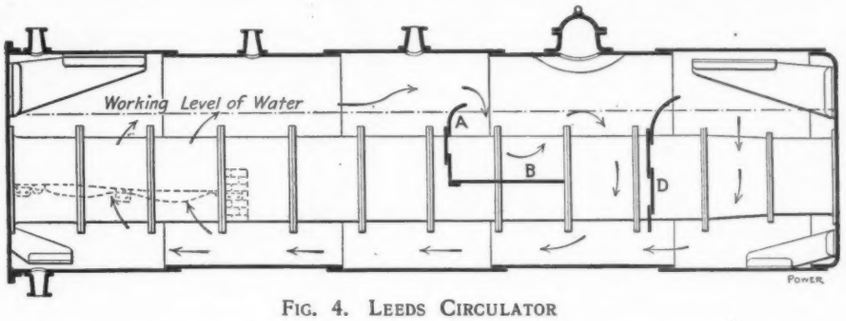


FIG. 4. LEEDS CIRCULATOR

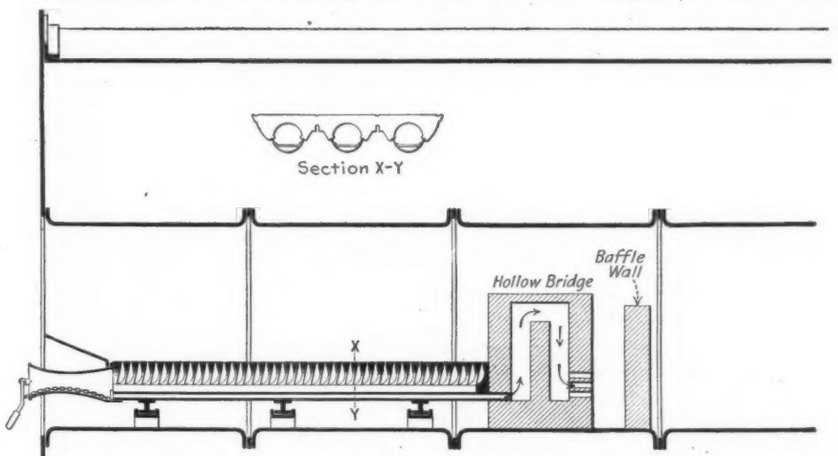


FIG. 3. WILSON AND FURNEAUX SMOKE CONSUMER

*This is similar to the steam loop used extensively in this country.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men

Recent Progress in Diesel Engines

By F. E. Junge

The efforts of Continental manufacturers of Diesel engines during the last few years have been devoted to some or all of three aims besides the attainment of utmost reliability of operation: The adaptation of the Diesel engine to the propulsion of sea-going vessels, the utilization of their waste heat, and the use of low-grade fuels, preferably such as can be procured in the home market. In order to attain the first aim it became necessary to create a high-speed type of engine of light weight. This was accomplished by reducing the piston stroke and therefore the hight of the engine, at the same time increasing the piston speed and therefore the power of the engine. The fear of designers that a high-speed Diesel engine would be inferior in heat economy to the old type was not confirmed in practice, the fuel-consumption figures obtained with the latter type being quite as good as those of the former.

Fig. 1 shows a high-speed Diesel engine of 300 horsepower built by Ludwig Nobel in St. Petersburg. The air is aspirated through side openings in the crank case, cooling the frame and the crank bearings on its passage to the cylinders. A vertical air pump is coupled to one end of the crank shaft and compresses air in two stages to from 50 to 70 atmospheres (750 to 1000 pounds per square inch). The governor is provided with an adjustment to vary the speed between 150 and 400 revolutions per minute; by reducing the quantity of fuel delivered by the pumps the speed can be still further reduced. The working cycle, the details of the steering gear, valves, etc., are the same, essentially, as in the older types of stationary Diesel engines and need not be here described.

The high-speed Diesel engines are lighter and less costly than the slow-speed engines and occupy less space. A high-speed four-cylinder engine of 250 horsepower is 11 feet 10 inches long, 39⅜ inches wide and 6 feet 11 inches high, while the corresponding proportions of the ordinary four-cylinder engine of the same horsepower are 13 feet 9½ inches, 78¾ inches and 10 feet 10 inches. The weight of the new type is 26,000 pounds against 50,485 pounds for the old type—both without the flywheel. The cost of a complete stationary plant, including cost of building, foundation and accessories, when equipped with the high-speed Diesel engine is 15 to 20 per cent. cheaper than one with the old equipment.

In the following are given some of the results obtained in a series of tests which were conducted some time ago on nineteen high-speed Diesel engines of from 250 to 300 horsepower capacity. The engines were all built after the same pattern, sixteen of them to drive

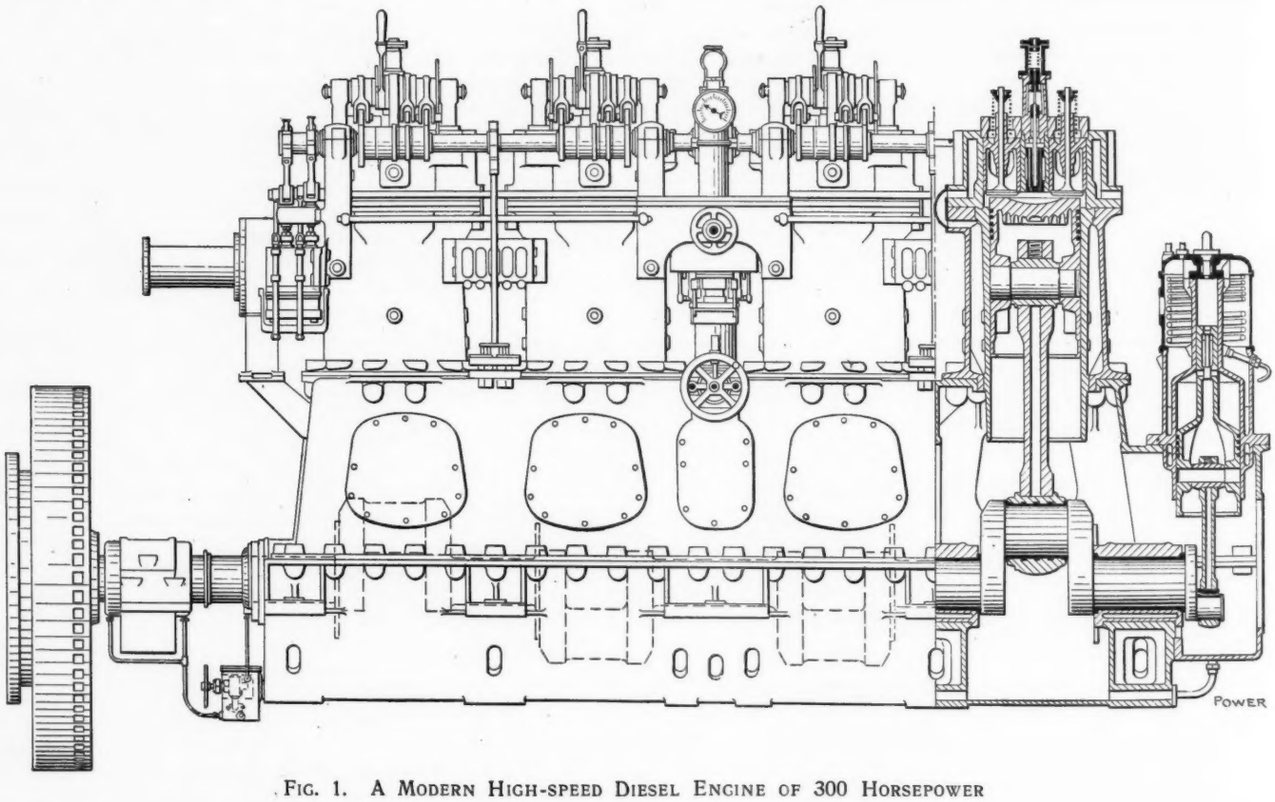


Fig. 1. A Modern High-speed Diesel Engine of 300 Horsepower

electric generators and three for ship propulsion; the latter were equipped with reversing gear. The tests lasted from 8 to 24 hours, the engines running under full load all the time. The piston speed varied between 14.7 and 16.4 feet per second. At the normal speed of 350 revolutions per minute the engines deliver 250 brake horsepower, and

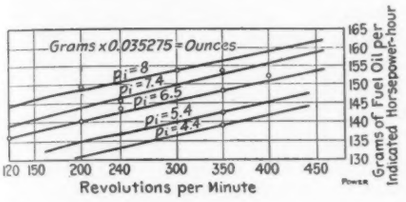


FIG. 2. RELATION BETWEEN SPEED AND FUEL CONSUMPTION

when forced they can deliver continuously 300 brake horsepower at the same speed.

The consumption of fuel oil remained within the limits of 185.5 and 205 grams (6.4 to 7.2 ounces) per brake horsepower-hour, at full load and 350 revolutions per minute, the exact figure depending upon the length of time which the engine had been in actual service before being tested. When a test is made immediately after erection, the fuel consumption is, of course, somewhat higher, owing to the higher mechanical resistances. On the average the fuel consumption per brake horsepower-hour at 350 revolutions per minute and overload is 205 grams (7.2 ounces), at normal load 195 grams (6.8 ounces), at three-quarter load, 215 grams (7.5 ounces), and at half load 235 grams, (8.2 ounces). The mechanical efficiency at full load averages 78 per cent. With slow-speed Diesel engines of the same capacity the consumption of fuel per brake horsepower-hour at full load is 195 grams (6.8 ounces), at three-

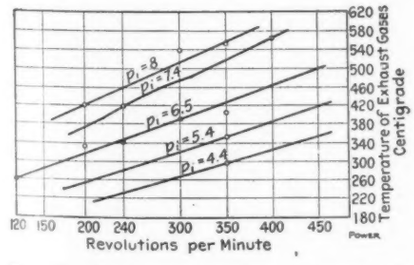


FIG. 3. RELATION BETWEEN SPEED AND TEMPERATURE OF EXHAUST GASES

quarter load 215 grams (7.5 ounces) and at half load 240 grams (8.4 ounces). It is evident that as far as the fuel consumption is concerned the high-speed Diesel engines are not inferior to the old type having a piston speed of 9.84 feet per second.

During the tests the engines worked partly with raw naphtha of a specific gravity of 0.88 (at 15 degrees Centigrade) and partly with solar oil of a specific gravity of 0.883; also with a mixture of 70 per cent. solar oil and 30 per cent. masut (specific gravity, 0.88), and finally with masut alone. Thus the high-speed Diesel engines were found to work on the low-grade fuels just as well as the slow-speed engines. As to the possibility of reversing, which is of prime importance for marine service, it was found that reversal from full speed in one direction to the other occupied from 9 to 10 seconds on the average. The speed of these engines can be reduced to 75 revolutions per minute.

Of the investigations of a four-cylinder engine which were made by Dr. M. Selliger and communicated to the Vereindeutscher Ingenieure, I give the following abstract: Speed, 350 revolutions per minute; diameter of working cylinder, 330 millimeters; piston stroke, 380 millimeters; diameter of low-pressure pump, 230 millimeters; diameter of high-pressure pump, 75 millimeters; stroke of pump, 180 millimeters. The analysis of the fuel naphtha showed 86.82 per cent. carbon and 13.17 per cent. hydrogen; calorific value, 19,575 heat units per pound. The relation between fuel consumption, pressure and speed is graphically shown in Fig. 2 and the change of temperature of the exhaust gases in its relation to pressure and speed is shown in Fig. 3.

The mechanical efficiency, by which is meant the ratio of the effective output of the engine to the difference between indicated horsepower and air-pump horsepower, decreases somewhat with the load, reaching its maximum of 80 per cent. at 300 revolutions per minute; at high speeds it is somewhat reduced. The heat balance of the process comes out as follows: About 40 per cent. of the heat contained in the fuel is transformed into indicated work; about 22 per cent. is absorbed by the cooling water; about 3 per cent. is lost through radiation; and the remaining 35 per cent. is carried away in exhaust gases and steam.

This is a summary of Doctor Seiliger's results: The fuel consumption per indicated horsepower-hour (135.5 to 154 grams) at constant mean indicated pressure decreases with decreasing speed of the engine. The fuel consumption per indicated horsepower-hour at constant speed decreases with decreasing mean indicated pressure (8 to 4.4 kilograms per square centimeter). The temperature of the exhaust gases (560 to 260 degrees Centigrade) at constant mean indicated pressure decreases with decreasing speed. The temperature of the exhaust gases at constant speed decreases with decreasing mean indicated pressure. The work absorbed in operating the air pump (6.4 to 20 horsepower) is directly proportional to the speed and independent of the mean indicated pressure. The negative work of mechanical resistances (36.1 to 82.3 horsepower) increases with the speed and with the increase of the mean indicated pressure. The proportion of cylinder volume filled with air (0.90 to 0.83) decreases with increasing speed (75 to 392 revolutions per minute) and is independent of the mean indicated pressure. (The upper and lower limits of the values obtained in the test are inclosed in brackets).

Doctor Seiliger found that an increase of the piston speed from 14.5 to 16.4 feet per second gave no increase in the capacity of the engine. When the number of revolutions per minute of the engine tested was increased from 300 to 350, the effective output of the engine increased from 263 to 304 horsepower. Between 350 and 400 revolutions per minute, which is an increase in piston speed of 15 per cent., the effective output rose only 3 per cent. Increasing the number of revolutions from 401 to 493,

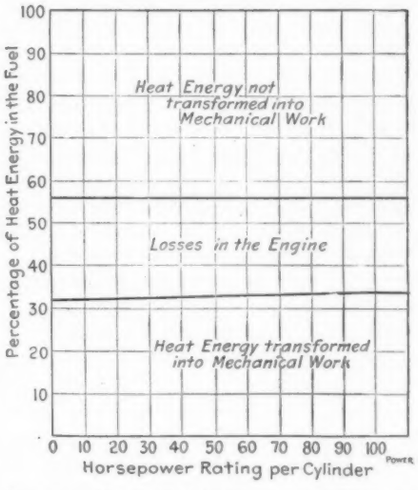

Heat Energy not transformed into Mechanical Work
Losses in the Engine
Heat Energy transformed into Mechanical Work
Percentage of Heat Energy in the Fuel
100 90 80 70 60 50 40 30 20 10
0 10 20 30 40 50 60 70 80 90 100
Horsepower Rating per Cylinder
POWER

FIG. 4. HEAT BALANCE FOR DIFFERENT SIZES OF ENGINE

which means an increase in piston speed from 15.7 to 19.6 feet per second, or 25 per cent., the effective output increased only by 30.5 horsepower, or 10 per cent.

UTILIZATION OF WASTE HEAT

Ordinary Diesel engines convert from 32 to 34 per cent. of the total heat contained in the fuel into available mechanical energy. A theoretically perfect machine could utilize 56 per cent., so that the economic efficiency of Diesel engines would then be 61 per cent. Fig. 4 is a diagram plotted by Professor Josse as a graphic presentation of the heat balance for different sizes of Diesel engines; Fig. 5, plotted by Professor Weber, shows the heat balance for various loads on a Diesel engine of 200 horsepower. Besides the losses designated as "energy not converted into mechanical work," part of the "losses within the machine" may be considered as capable of utilization, because a considerable portion of

the friction work is transformed into heat and added to the cooling water. On the other hand, there are irretrievable losses of heat through radiation from the machine to the surrounding atmosphere, so that a certain portion of the distances between the curve *A B* and the line *C D*, depending on the momentary load, must be considered as the percentage of waste heat which is available for other purposes.

As regards the cooling water it was found that in Diesel engines with complete utilization at least 500 heat units per brake horsepower-hour can be reclaimed, no matter what the size of the engine or its load. It was more difficult to ascertain the exact amount of heat which can be reclaimed from the waste gases. A waste-heat economizer built by the firm of Sulzer in Winterthur, Switzerland, utilized from 10.5 to 17.8 per cent. of the total exhaust heat while the cooling water absorbed between 28.2 and 30 per cent., so that, on the whole, from 171,000 to 264,000 heat units were utilized, the amount varying with the arrangement, namely, whether the engine and the economizer are arranged in parallel or in series. Figuring the money value of these savings, realized every day in the year, it is found that the initial cost of the apparatus for utilizing the waste heat is insignificant compared to the economies gained. The economizer was arranged somewhat in the fashion of sectional book cases, one section being placed on top or alongside of the other and the number of sections depending upon the amount of heat and space available.

Similar apparatus are used to raise low-pressure steam by means of waste gases, though, for the latter purpose, Diesel engines are not so well adapted. All the required heat must be furnished by the exhaust gases. Hence, even if, instead of new cooling water, the condensate is returned at some 90 degrees Centigrade to be used over again, the steam-raising capacity of the plant is comparatively small, though it is sufficient to supply a secondary steam-heating plant, or to distil water for chemical purposes or for filling storage batteries.

Tests made by the firm of Sulzer on a 150-horsepower plant showed that at least 400 heat units per brake horsepower-hour can be regained from the exhaust gases. This rate, as well as the rate of heat recovery from the cooling water, increases with decreasing load. Hence, from the exhaust gases and cooling water combined, at least 900 heat units can be recovered per brake horsepower-hour at normal loads, and under especially favorable conditions the total amount recoverable may be as much as 1500 or more heat units per unit of output. The division of this total amount between the cooling water and the exhaust gases varies somewhat according to the mode of operation.

As to the heating surface required for waste-heat accumulators, 2⅓ square feet per brake horsepower are found sufficient, even when taking into account the gradual settling of sediment on the inner surfaces. This ratio is given with the provision that the waste gases yield their heat to water and on the counterflow plan. High water and gas velocities and long contact surfaces are favorable to heat transfer, the velocity of the gases being of far more importance than the velocity of the water.

When the engine and the economizer are arranged in series instead of in parallel, it is advisable, owing to the smaller temperature difference between gases and water, to make the heating surface larger than 2⅓ square feet per brake horsepower-hour, in order

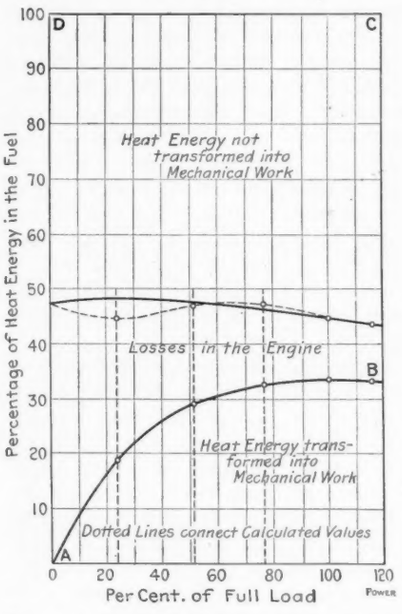


FIG. 5. HEAT BALANCE FOR DIFFERENT LOADS ON THE SAME ENGINE

to be sure that the exhaust gases will be cooled down from their normal temperature (200 to 500 degrees Centigrade according to load and size of the engine) to 100 degrees Centigrade, even after the walls of the economizer have been coated with sediment. For heating air, owing to the lower coefficient of heat transference, the heating surface must also be made considerably larger. It should be added that the waste-heat economizer imposes no excessive back pressure on the engine; the back pressure is not greater than that caused by the ordinary exhaust piping, while the economizer also acts as a muffler, making the exhaust absolutely noiseless.

UTILIZATION OF LOW-GRADE FUELS

The control by the Standard Oil Company of the world's trade in oil products has forced the German engineer and chemist to find a substitute for the trust products which will permit them to satisfy the demand for oils in industrial and other pursuits from native resources, if the prices of oils should be raised unreasonably or if for any reason the supply of foreign oils should be cut off. This substitute has been found in the by-products of the gasification of coal in gas-house retorts, coke ovens and gas producers. In 1909 the total annual capacity in these coal oils was 90,000 tons from German brown coals and 300,000 tons from hard coals. Altogether, 400,000 tons of oils can be produced from German native coal resources per annum, if the necessity should arise. Of the hard-coal oils, anthracen and creosote oils are used in Diesel engines, the consumption being about 6.3 ounces per brake horsepower-hour for oil of 18,000 heat units per pound, costing about $1 per 100 kilograms (220 pounds).

Recently, the firm of Körting Brothers has made successful attempts to use tar as fuel for Diesel engines. For starting the engine paraffin oil is used, both fuels being delivered to the engine by special pumps. The total heat consumption, if both fuels are used, is approximately the same as when oil alone is used. From tests recently made it was found that a normal Diesel engine of 100 horsepower which ordinarily uses 7340 heat units per horsepower-hour consumes the following quantities of tar and oil: at full load, 7.46 ounces of tar and 0.13 of oil; at three-fourths load, 7.04 ounces of tar and 0.48 of oil; at one-half load, 6.62 ounces of tar and 0.84 of oil. The low heat value of the oil was 18,000 heat units per pound and that of the tar 15,300 heat units per pound. The tar used in the tests came from several gas houses. In two cases it was a byproduct of English coal; in one case, of Westfalian coal, and in two cases, of coal from upper and lower Silesia.

Beside the fuel-consumption test a duration test of 66 hours at two-thirds load was made. Upon examination of the cylinder no residuals from the combustion of the tar were found, nor was there any sediment on the valves or the nozzles. In point of speed regulation it was found that the tar gave the same satisfaction as the oil fuels. In view of these favorable results the Körting firm has given orders for two 600-horsepower horizontal Diesel engines to be used in the central station in Dessau, both to run on gas-house tar.

Old Sandy McPherson and young Aleck McDonald were working together over at the power plant, when Aleck says, "Sandy, what is this 'ere stuff they call vaakum?" Sandy replies: "It is naught, lad; it is nothing." Aleck says: "Sure it must be som'ot, Sandy, it must be something, for they keep it in pipes here."

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment

Using a Dynamo Electric Machine Interchangeably as a Generator and as a Motor

By C. C. Hoke

Conditions sometimes arise in power plants which render it desirable to operate a compound-wound machine either as a generator or as a motor, absorbing energy from a line shaft and delivering electrical energy in the first case, or *vice versa*. This situation has been met with by the writer on two occasions.

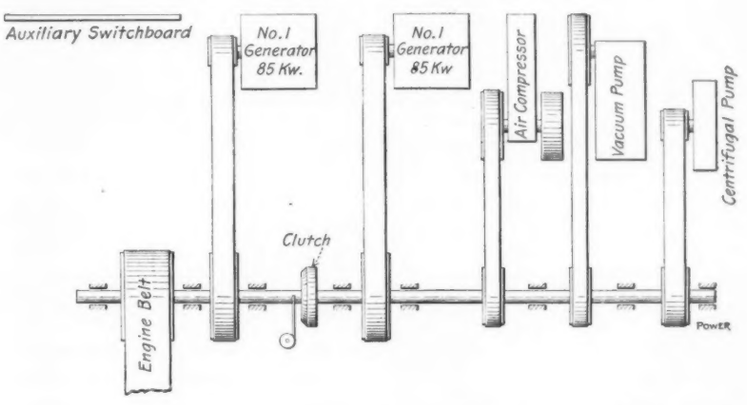


Fig. 1. Arrangement of Drive for Small Generators and Pumps

In the first instance a plant was normally supplied with electricity by means of a 250-kilowatt direct-current generator. Two 85-kilowatt machines were installed for relay service, and these were belted to a line shaft which was driven by a simple engine and which drove a belted air compressor and the condenser auxiliaries for the entire power plant, as shown diagrammatically in Fig. 1. A friction-clutch coupling was inserted in the line shaft between the pulleys driving the two small dynamos and by means of this clutch No. 1 generator and the compressor and condenser auxiliaries could be disconnected from the rest of the equipment, if desired. Under normal service conditions this was done, the disconnected section of the shaft being driven by the No. 1 machine operating as a shunt-wound motor and taking current from the busbars connected to the 250-kilowatt machine. The conversion of No. 1 auxiliary machine from compound to shunt-wound was accomplished by means of the switching apparatus and connections shown in Fig. 2. The two separate switchboards were used on account of the desirability of controlling the No. 1 machine when operated as a motor from a point near the condenser auxiliaries and as a generator from the main switchboard.

The reason for using the field rheostat when operating as a motor was to bring the machine up to the normal speed and also to secure some speed regulation, this being desirable under the varying load demands. At times the purely mechanical equipment only was driven by the engine, letting the generators run idle, but this was not often the case, being done only when the load happened to be too high to be carried entirely by the 250-kilowatt unit. At times the engine drove the mechanical auxiliaries and one 85-kilowatt machine in multiple with the 250-kilowatt unit and frequently the two small generators were run in multiple at light load, carrying all of the electrical load, in order to make repairs or adjustments on the larger unit. In this case the centrifugal pump was usually relieved by a steam pump and its belt was slipped off, in order to permit throwing all the load possible on the small generators up to the capacity of the engine. The compressor was likewise run intermittently, being connected to the main shaft through the medium of a friction-clutch pulley.

The second case was substantially the same as the one just described except that a No. 7 Connersville blower entered into consideration instead of the air compressor, etc.

This idea may no doubt have been applied by others but it may be new and of value to some readers, in which case the writer will feel amply repaid for presenting it.

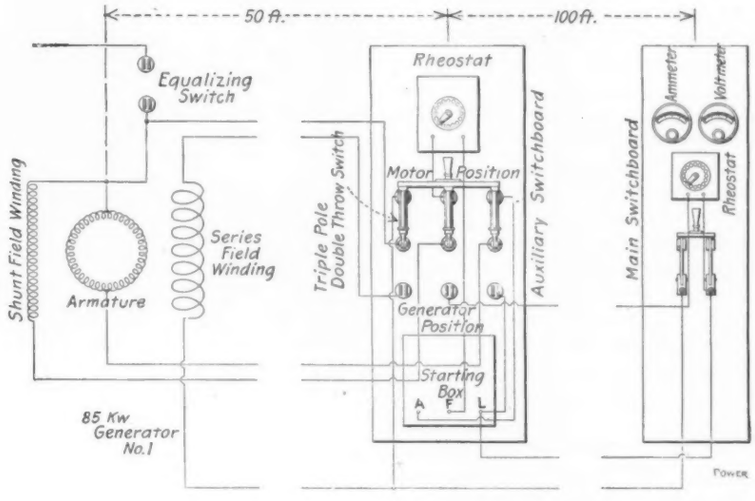


Fig. 2. Diagram of Switchboard Connections

Maintenance of Railway Motor Bearings

By C. J. Fuetterer

For any electric-railway company operating a lot of equipments of the same type, it is almost a necessity to keep bearings in stock, ready for use. Since the journals on the motor shafts wear down more or less after a certain length of time, however, it would be impractical to bore all the bearings the same size, and the best plan to follow is to keep in stock bearing bushings standardized, say, to three sizes differing slightly in diameter; the large size to be used on new equipments, the second size for equipments in which the journals have worn enough to justify turning them down to the diameter necessary to take the second standard size, and the next or largest size of bushing to be used in a similar way for equipments requiring a further truing up of the journals.

In boring the sleeves, allowance should be made, in order to take care of oil and unavoidable irregularities, of 0.002 inch per inch diameter of bearing for solid sleeves and 0.003 inch per inch diameter of bearing for split sleeves.

Maintaining Voltage by "Forced Draft"

By Hudson R. Biery

The Indianapolis & Louisville Traction Company, which was a pioneer in adopting 1200 volts for direct-current railway operation, has applied an interesting expedient for maintaining the full rated voltage of the generators at its Scottsburgh, Ind., power plant under very unfavorable conditions. The generators are General Electric 600-volt 600-kilowatt machines, with two armatures mounted on a common shaft and connected in series in order to deliver the 1200 volts on which the cars operate. As the road has no substations and as the termini of this division are each 20 miles from the power plant, it is essential that the voltage be held as near 1200 as possible. The machines were designed to run at 120 revolutions per minute, but owing to a misunderstanding in the construction of the plant equipment, it was necessary to put the speed down to 115 revolutions per minute. At this speed the voltage would range from 1180 to 1160, starting at the first figure in the morning when the load was light and the field-magnet coils were cool, and gradually dropping to the latter figure as the field windings began to warm up.

It was decided that about the only inexpensive way in which to obtain the desired voltage was to keep the field windings cooler and, accordingly, a blower outfit was installed. A small 5-horsepower motor running at a maximum speed of 1200 revolutions per minute was belted to a 6-inch fan which has a capacity of about 2000 cubic feet of air per minute. An intake pipe 10 inches in diameter brings air to the fan from outside the building. The motor is equipped with a regulating rheostat which provides two speeds, the low speed being used in the morning when the load is not heavy and the field-magnet coils are cool, and the maximum speed being used when the voltage begins to drop, as the coils warm up. A 10-inch galvanized delivery pipe extends from the blower under the floor to the generators.

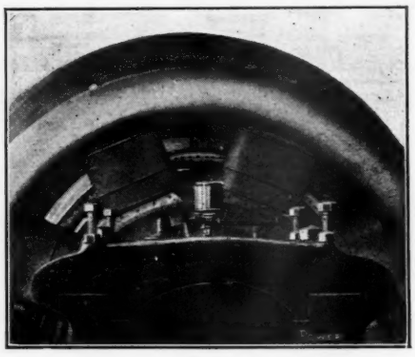


Fig. 2. View of Pipe through Machine Frame

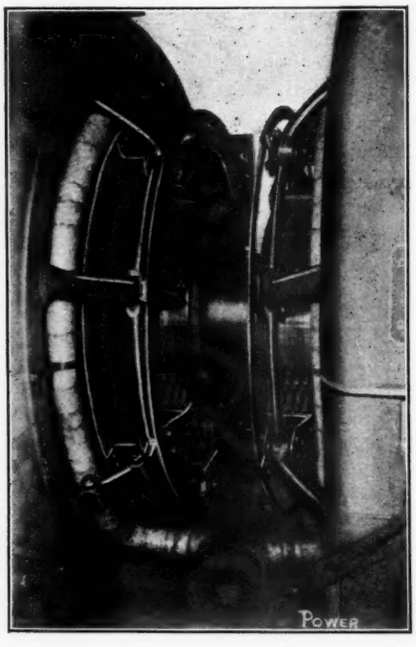


Fig. 1. Arrangement of Coil-cooling Air Pipe

At the point where the pipe emerges from the floor between the generators, the size is reduced to about 3 inches and divided into two branches. Each of these branches entirely encircles the armature of one machine, as shown in Fig. 1, and a row of holes ½ inch in diameter and 2 inches apart in the wall of the pipe next to the machine delivers jets of air directly on the field-magnet coils. Fig. 2 is a view from the other side of one of the generators, showing the perforated side of the pipe.

With the blower in operation it is now possible to maintain a voltage ranging from 1240 to 1260 without increasing the temperature of the field-magnet coils beyond the danger point.

Although the idea of using a blowing outfit to keep down the temperature of a generator is not new, it is probable that the machines here described are the only ones of their type to which such an installation has been applied. The system was installed under the direction of H. D. Murdock, superintendent of the road, and is under the supervision of chief engineer Wesley Hartley.

CORRESPONDENCE

Hydroelectric Expansion in California

According to the daily papers, the Pacific Gas and Electric Company plans to develop an additional 71,000 hydroelectric horsepower at a cost of about $10,000,000. This will make the company's total capacity 260,000 horsepower. The company now serves about 38,000 square miles in central California.

The present project is said to provide for the erection of a dam in the cañon of the South Yuba river, and the construction of two power houses, the first to develop 50,000 horsepower and the second, which will use the water again, to have a capacity of 21,000 horsepower. The overflow of water is to be used for irrigation purposes and additional acreage is being developed in Placer county for fruit-growing purposes.

Operating Water-driven Alternators in Parallel

I note in the issue of June 27, an inquiry from Mr. Dean relative to the operation of small alternating-current water-driven plants in parallel. The accompanying diagram covers the system which he outlines and the arrangement as laid out will be found very convenient and flexible. I have used the simple single-line diagram for the system which can be easily filled out for three-phase circuits. Upon analysis the reader will find that rather free use of disconnecting switches has been made; while this may be a little extravagant for so small a system, I strongly recommend that they be installed, because one cannot estimate their value in cases where it might mean the crippling of the entire system for one small mishap to some particular switch or other equipment. I also recommend the tie, shown by the dotted line, between the transmission line and the outgoing feeder, as it would

enable the two small units to carry load even though the larger plant was out of service.

I advise the use of station busbars at the largest plant in order to give flexibility and to have a means of knowing what amount of the load is being carried by the two small plants.

It would seem best, from what information was given in Mr. Dean's letter, that the instruments be rearranged at the 200-kilowatt station as sketched and that the switchboard be made up of three main panels, namely, one general panel, one incoming line panel and the feeder panel; an exciter panel will be necessary also, if the exciter instruments are not carried on the general panel, and I would advise the use of a swinging bracket to carry the synchroscope, frequency meter and voltmeter. Voltmeter receptacles should be mounted, one on the general panel, one on the incoming-line panel and one on the feeder panel. The synchronizing receptacles should be arranged in the same order at the main (200-kilowatt) station, but at the small stations it will be necessary to use only one plug, although two could be used. I have not shown any station busbars at the two smaller stations, as they will act only as simple generating stations feeding one line.

There has been hardly enough information given to enable one to advise intelligently as to the use of governors at the two smaller stations. If the load on the system is subject to rather sudden variations and liable to have a minimum load of less than 100 kilowatts at such times, it will be necessary to have a governor on the 60-kilowatt unit at least. It is quite common to run the prime movers of several alternators of an installation "wide open" and let the others do the governing, although this impairs the regulation to a considerable extent. If the load on Mr. Dean's line is such that he can always rely upon at least 90 to 100 kilowatts, barring the opening of circuit-breakers, it is quite feasible to use but one governor, located at the main generating station; but I would advise the installation of a governor also at the 60-kilowatt plant.

In regard to the actually necessary instruments that will be required, I would say that all those indicated on my diagram will be found very useful, but it is not essential that wattmeters be installed nor that a frequency meter be used; also, the synchronizing can be done with lamps, but I would not consider that good modern practice.

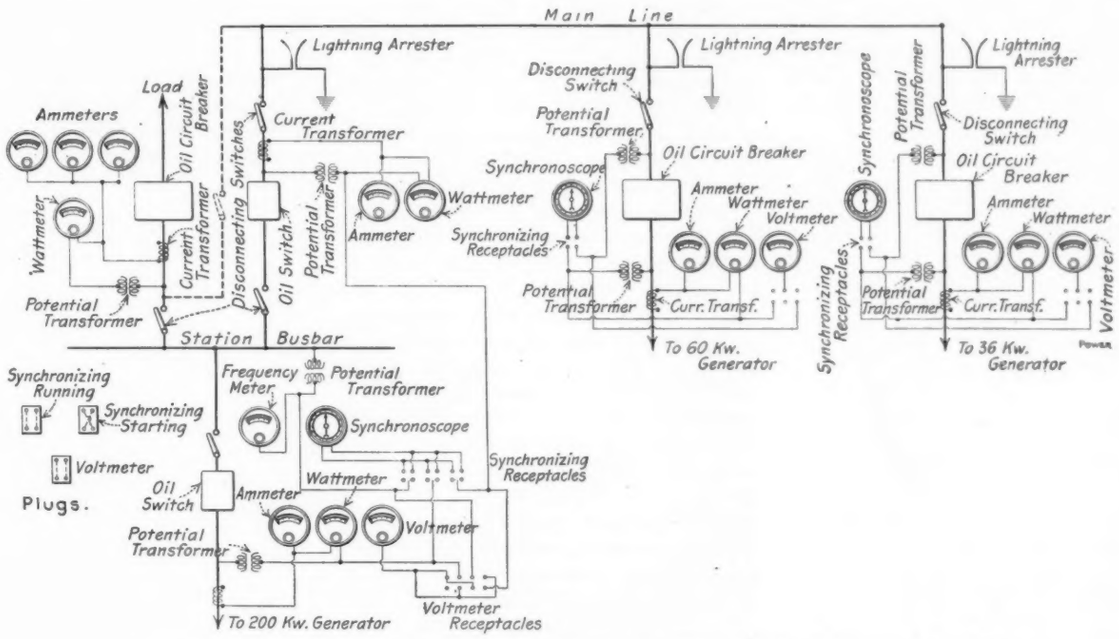
Throwing the machines on the line is a very simple matter. Build up the voltage on the incoming machine to that of the system, then put in the synchronizing plug or plugs, as the case may be, and the synchroscope will indicate whether the incoming machine is running fast or slow. Regulate the speed of the turbine until the synchroscope pointer moves very slowly, which indicates a very slight difference in speed. It must be remembered that changing the speed of the incoming machine will affect the voltage of the machine and that it must be adjusted to equal that on the system before closing the switch. As the synchroscope hand approaches the mark, close the generator switch just an instant before the hand shows synchronism and the machine should go into parallel just as smoothly as a direct-current machine would.

The operator must be careful to insert the synchronizing plugs in the correct receptacles; the "running" plug must go in the busbar or line receptacle and the "starting" plug in the receptacle on the panel of the incoming generator; if the generator is already supplying energy to the system, however, and it is desirable to "phase it in" on the line or the busbars, the running plug must be put in the generator receptacle and the starting plug in the line or busbar receptacle. After the machine has been connected to the line, simply open the turbine gates until the desired load is on the machine. The field current should be adjusted to the same value as if the machine were not in parallel, for the same individual load.

When it is desirable to shut down any machine, simply close down the turbine slowly, gradually cutting down the field current of the machine also, until the wattmeter (or, if there is none, the ammeter) shows no load; then open the switch.

I should be pleased to hear further particulars of the installation from Mr. Dean, either through POWER or personally.

EDWARD L. NUTE.
Master Mechanic, Connecticut River Power Company.
Vernon, Vt.



ARRANGEMENT OF INSTRUMENTS AND CIRCUITS FOR MR. DEAN'S PARALLELED STATIONS

Electric light and power in all of the larger towns of the Willamette valley, Oregon, was shut off a few days ago as the result of a fire which destroyed the plant of the Kelly Lumber Company, one of the largest inland sawmill concerns in the Northwest.

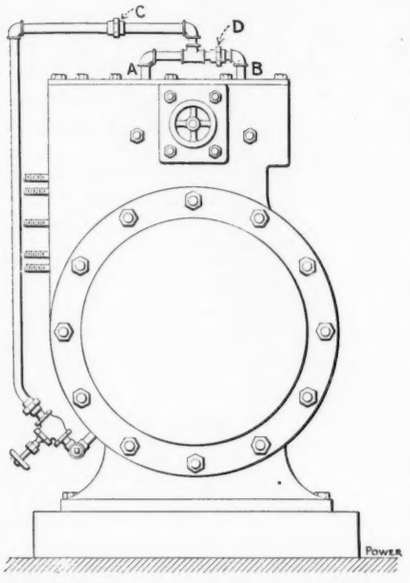
Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words wanted

Air Discharge Valves Cooled by Water

Trouble is had frequently with lubricating oil burning on the valves of the air end of dry-vacuum pumps. The illustration shows how I have overcome this trouble on my air pump, by the introduction of cooling water at the discharge side of the valves.

In order to get cold water I put a tee in the supply pipe of the air cylinder-cooling jacket, about midway of the cylinder, and, with a valve and fitting ran a ¼-inch pipe from the tee up between the suction and discharge pipes, over the top of the valve chest, as shown



AIR-DISCHARGE VALVE COOLED BY WATER

at *A* and *B*. The two unions, *C* and *D*, are for convenience when removing the valve-chest cover.

The first pump on which this scheme was tried had given considerable trouble after running less than a week. The valves and valve seats would coat with burned oil and would drag so badly that a shutdown and cleaning would be absolutely necessary. Large quantities of oil would sometimes stop the dragging for a few moments, after which it would be worse than ever.

Both the valve and valve seat were badly cut. That the pump might run more than 24 hours without dragging, paper liners were used under the valve plate, thus giving the main valve about 0.004 inch more clearance.

At the time water was piped to the discharge valves of this pump it was shut down because the valves would drag, but they had not been cleaned. After connecting up the water I started the pump with a full pipe of water flowing onto the valves. As the burned oil made the valves fit tightly a vacuum of between 29 and 29¼ inches was had at the start.

After running about two hours, however, the vacuum again went back to 22 inches, and the pump was stopped and the valves examined. The water had washed out the burned oil from between the valves and their seats and also from the score marks before mentioned, which allowed considerable leakage. This was remedied in part by removing the paper liners under the valve plate, thereby reducing the clearance between it and the valves.

The pump was again started without water, but with a little oil, which was shut off after running a short time, and allowed to burn until it had filled the score marks and other leaks enough to bring the vacuum up to 29⅔ inches, when the water was again turned on, but in a smaller quantity.

After much experimenting the right amount of water was determined (in this case the water valve was little more than cracked open) to keep the valves from dragging and still maintain the maximum vacuum.

The pump has now been running about three months without a cleaning. It is pulling a better vacuum and is running smoother with less steam on 50 per cent. less oil than it ever did before.

As the right amount of oil and water is admitted to the valves, it is churned to a lather, which lubricates them perfectly. I believe a very low-priced oil would give satisfactory results.

It only required a little water in the right place, just enough to prevent excessive friction.

W. E. BERTRAND.

Philadelphia, Penn.

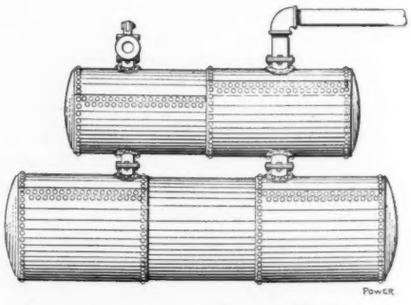
Steam Drum to Prevent Wet Steam

A cement plant was started up in February, 1910, and a great deal of trouble with the boiler-feed water has been experienced ever since.

Rio Grande river water is used, and as this river only runs three or four months during the year, a reservoir, in which is stored the year's supply of water, has been built about one mile above the plant At certain periods of the year when the river is running, the water is fairly good, carrying about 20 grains of solids to the gallon. During these periods analysis is made of the water every day and it is pumped into the reservoir when the total solids are 20 grains or less. Due to evaporation during the summer, the water in the reservoir gets more concentrated and, in the latter part of the season, runs as high as 50 grains to the gallon. The following is the analysis of a concentrated sample:

	Grains per Gallon
MaCl	282.68
Na_2SO_4	13.69
$CaSO_4$	49.03
$MgSO_4$	18.14
$MgCO_3$	13.26
$FeCO_3$	0.530
SiO_2	0.610
Volatile and organic matter	21.91
	399.85

While the water carries a great deal of scale-forming matter, no trouble is had. As there are four 400-horse-



ARRANGEMENT OF STEAM DRUM TO ELIMINATE WET STEAM

power water-tube boilers in the power plant it is only necessary to run two of them at a time, which gives ample opportunity to clean out and make any necessary repairs on the boilers not in service. The great trouble is with the boilers priming, which no doubt is caused by the amount of calcium chloride which gradually increases as the water gets

more concentrated day by day. Water is carried in the boilers as low as is safe, but still at times the boilers prime, and in a number of cases this has shut down the plant. The boilers are equipped with superheaters and serve two 750-kilovolt-ampere turbines.

Various people who make a specialty of treating water have taken up the matter, but none of them seems to be able to remove the cause of priming. The dry pipes have been remodeled and the boilers equipped with skimmers, but to no avail.

I would like to have the opinion of some of POWER's readers as to whether my scheme will overcome this trouble.

To the center drum of each boiler there is attached a 4½-inch safety valve, and a 6-inch saturated-steam line which carries the steam through a U-bend to the superheater header. My idea is to mount a 36-inch by 10-foot steam drum on each boiler, putting flanges on the drum diametrically opposite the connections to the boiler for the safety valve and the saturated-steam line, as shown in the illustration. A drum can be put on each boiler when it is out of service by simply removing the U-bend and safety valve and remounting the safety valve on the flange directly over its old position. The addition of this drum would increase the steam space and the area of the steam opening from the drum on the boiler into the auxiliary drum equal to the area of the 4½-inch safety-valve opening.

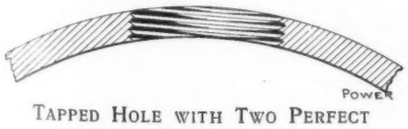
I think this scheme would overcome the priming, but as the addition of this drum would be quite expensive, I would like to have the opinion of others before making the change.

L. D. GILBERT.

El Paso, Tex.

Direct Branch Pipe Connection

It is often necessary to tap directly into main pipe lines for the connection of minor branches. Although a standard threaded tee or fitting is the correct method of connecting branch pipes, direct tapping into the pipe is permissible within certain limits.



TAPPED HOLE WITH TWO PERFECT THREADS

This method of connecting should never be employed in case of a drain, or where considerable vibration occurs, or where bending is imposed upon the branch by the expansion and contraction of the main. The thinness of the pipe walls offers but little opportunity for a substantial joint and the curvature of the pipe greatly increases the difficulty. The effect of the curvature upon the continuity of the thread in contact is shown in the accompanying illustration.

The accompanying table shows the smallest mains that will accommodate branch pipes and the number of perfect threads. In the mains of the larger sizes more substantial connections could be made, due to the more gradual curvature and thicker gage of pipe.

DIRECT TAPPING FOR BRANCH PIPE

Diameter of Branch Pipe	Diameter of Smallest Main Pipe		
	2 Perfect Threads	2½ Perfect Threads	3 Perfect Threads
¼	2	1	1¼
⅜	2	2½	2½
½	2½	2½	2½
¾	3	3	4
1	4½	3½	4
1¼	6	7	8
1½	6	8	9
		8	10

Use combinations equal to or better than those given above the heavy line.

The number of perfect threads given is the number of threads in continuous contact in the branch joint, and as they have a direct bearing upon the tightness of the joint, the larger the number or the nearer the standard the threads are, the better. The number of threads given is hardly more than one-third of the standard; however, the joints represented above the heavy line give very satisfactory results.

The heavy line distinguishes the joints in which the strength is equal to or greater than that of the branch from those in which the strength of the branch is greater than the joint. In the joints represented below the line the strength of the branch pipe predominates so decidedly over the strength of the joint that their use is not advisable.

J. W. TAYLOR.

Massillon, O.

Installing Oil Tanks

I am installing two oil tanks for storing fuel oil to be used in forging furnaces, and expect to place them about five feet from the ground.

Will some reader of POWER who has had experience give me an idea of a cheap and efficient arrangement for transferring the oil from the tank cars in which it is shipped to the storage tanks mentioned? The tank from which the oil is to be taken will be about 3 feet below the storage tank. It should also be borne in mind that oil is somewhat more difficult to handle in freezing than in warm weather.

W. W. WARNER.

Kent, O.

Tarred Paper Gaskets

When an air or oil joint persists in leaking, try a gasket made of tarred paper. I have been using tarred-paper gaskets on the flanged joints of two Lombard governor pumps for two years and find them much better than the lead gaskets.

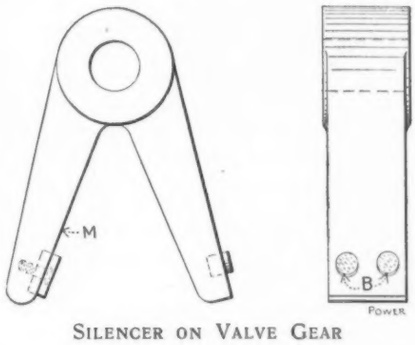
I am also using tarred-paper gaskets on the flanged joints of an air compressor with excellent results.

LEROY D. WHITE.

Oldtown, Me.

Making Corliss Valve Gear Noiseless

The accompanying illustration shows a method of rendering some forms of Corliss engine valve gears practically noiseless. I have taken valve gears which were quite noisy and made them almost silent in operation by drilling two holes in the crab claw, as shown at *B*, and inserting pieces of leather in them.



SILENCER ON VALVE GEAR

For some gears it might be better to insert the leather at the point shown at *M*.

C. R. MCGAHEY.

Baltimore, Md.

Emergency Oil Controller

Two years ago there was installed in a certain electric-light station a large high-speed, cross-compound Corliss engine, having a generator mounted on the same shaft with the flywheel, both revolving between two main bearings. The low-pressure steam eccentric is between the generator and the flywheel, which are only 10 inches apart, making it impossible for the attendants to feel of the eccentric while running.

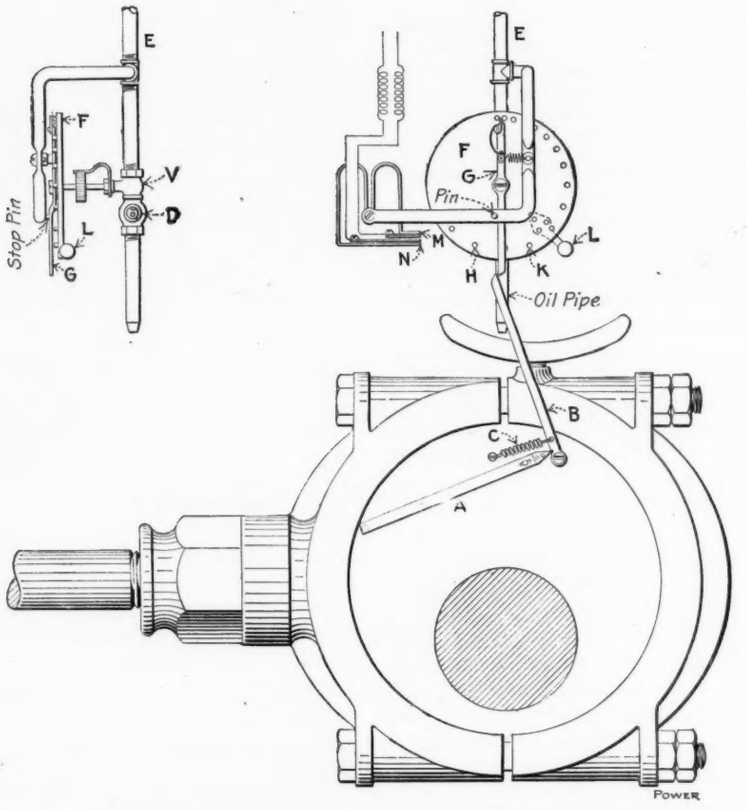
One night the eccentric became so heated that the straps had to be rebabbitted. When repairs had been completed, attempts were made to invent some protective scheme that would sound an alarm and automatically admit more oil, if the eccentric again was unduly heated. Experiments based on the fusible temperature of various wax com-

pounds and metal alloys proved unreliable and troublesome, and were finally relinquished in favor of a plan evolving the principle of the expansion of metals by the increase of heat.

I believe the arrangement adopted may be readily modified to become applicable to almost all kinds of unreachable bearings as, for instance, the outboard journals of engines having their flywheels set so close to the engine-room wall as to block sufficient passage to feel of them.

Referring to the sketch, *A* is a strip of aluminum anchored at the blunt end and fitted with a sharp steel tip at the other, which engages in a recess filed in the lever *B*. The spring *C* is stiff altogether for a portion of the circular arc to the left.

Therefore, when the aluminum strip *A* expands a trifle, due to a slight rise of temperature in the metal of the eccentric, the lever *B* will engage with the lever *G* and the disk *F* will be turned an amount depending on the length of contact between these two levers; thus opening the valve *V* and causing the oil to drip slightly faster at the sight feed *D*. When the eccentric becomes so heated that the lever *G* swings enough to allow the pawl to engage the pins, which are the greatest distance apart, as *H* and *K*, then the disk will move at every stroke and the ball *L* (now being on the other side of the disk) will



SHOWING DETAILS OF OIL CONTROLLER

enough to overcome the centrifugal force of the lever *B* when the eccentric is in motion. It is apparent from the relative positions of the two levers that slight changes in the length of *A* will cause quite a movement in the top end of *B*.

The system of lubrication is of the gravity type, the oil descending from the upper tank to the sight feed *D* through the pipe *E*. The common handle of the controlling valve is replaced by the pin disk *F* and loosely fitted on the same stem is the pawl-ended lever *G* which engages with the small pins shown. The distance between these pins gradually increases for the first quarter of the disk and the pins are missing descend on the electrical spring contact plates *M* and *N* and cause an alarm to be sounded.

The attendant will then watch the eccentric closely until it either cools down from the extra supply of oil or shows indications of smoking.

M. CASSIDY.

South Framingham, Mass.

Steam in Cold Boiler

Some years ago I was employed in a plant the equipment of which included two Babcock & Wilcox boilers, carrying 100 pounds steam pressure. The tube caps had not been removed for a number of years.

A tube cleaner was procured and the work of cleaning the first boiler began. When it came to replacing the caps some difficulty was experienced in making the joints tight; consequently it was necessary to reface some of them several times, testing them under pressure each time.

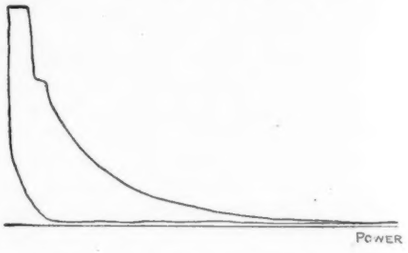
The chief decided that instead of firing up the boiler each time it would be quicker and easier to fill the boiler with cold water to the normal running level and then admit steam from the other boiler through the main stop valve and bring up the pressure in that way, which was repeated several times, and nothing happened.

LEON ROUNDY.

Concord Junction, Mass.

Indicator Diagram

Will some steam-engine expert tell what is the matter with the engine from which the accompanying diagram was



taken? Six different indicators were used with the same result.

H. T. FRYANT.

Jackson, Miss.

Leaky Water Tanks

Most engineers have had more or less trouble with leaky wooden tanks at some time or other. I refer to tanks that, either through carelessness or a shortage in the water supply, have been allowed to dry out some distance down from the top.

When a tank once gets in this condition it is usually difficult to reclaim it. Some use bran, others sawdust, oakum and tar.

A method that I have used consists of putting narrow strips of cardboard into the cracks between the staves.

This is a somewhat tedious procedure, but the moment the water reaches the cardboard it swells and stops the leak. It will also give way to the wood as soon as it begins to swell and finally decompose, leaving the tank tight with the staves in their former condition.

All of the hoops should be pulled up tightly before commencing and when the tank is tight it should be given as many coats of boiled linseed oil as it will absorb and then finished off with a coat or two of some good paint.

EARL PAGETT.

Coffeyville, Kan.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

The Cornell Economizer

I was somewhat amused to read the editorial in the July 4 number on "The Cornell Economizer," because an examination of this device appeals to me as similar to trying to lift yourself by your boot straps. The only possible gain that can be accomplished by a device of this character is draft, and it is very doubtful if the cost of repairs and replacement in this device would warrant the installation of any such device for improving the draft. It would take but a very small increase in the size of the stack or the installation of a very small forced-draft fan to accomplish the same result, with less wear and tear, less first cost and less operating cost.

In the two installations coming to my notice where these have been installed there has been an increase in the coal burned and the steam generated, but it has cost approximately $300 a year for replacement of the retorts in each boiler, besides the labor and cost of the steam required to operate them. A very slight change in the arrangement of the boilers, viz., increasing the flue connections, would have accomplished the same results at a first cost of not over $60, and thereafter no operating cost whatever, as the stack in both cases had ample capacity to handle the boilers. The trouble was due to poor setting and constricted flue connections between the boiler and the stack.

HENRY D. JACKSON.

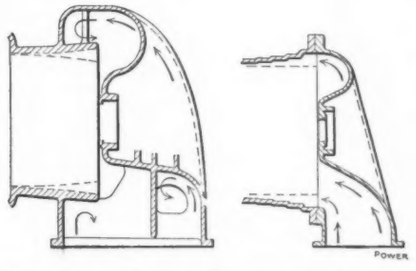
Boston, Mass.

Design of Turbine Exhaust Outlet

Among the many good points of the turbine design shown in the figure on page 987 of POWER for June 27, which is one of the illustrations of the excellent article of Messrs. Junge and Heinrich on "The Steam Turbine in Germany," one in particular deserves marked attention, that is the exhaust end of the machine which is reproduced here in Fig. 1.

Designers are not agreed on the most desirable arrangement of the turbine and the condenser. Some wish the one placed directly over the other, so that the exhaust steam and the large proportion of water it contains may flow directly and with the least possible resistance into the condenser; which is sound, and particularly so in view of the benefits resulting from the maintenance of a high vacuum. Others would rather sacrifice ½ inch or 1 inch of vacuum in order to accommodate a given type of condenser. In many instances the steam is led from below the turbine up through a large pipe 30 or more feet high, thence through a return leg connected to the condenser proper. With this arrangement, in case of failure of the air pump, the water cannot back up and flood the turbine.

While turbines are usually well guarded against the danger of flooding, large units depending upon siphon pipes and small ones upon nonreturn valves, the records show that quite a number of



FIGS. 1 AND 2. TWO DESIGNS OF TURBINE EXHAUST OUTLET

machines have been seriously damaged through the impact of water coming back through the exhaust end.

Invariably the trouble happens thus: the machines running with their usual smoothness are suddenly and violently shaken up; the single-disk turbine becoming unbalanced, contact takes place at the safety bearings and the motion stops; in multistage turbines there is much grunting and grinding, and sometimes, the trouble being apparently over, the motion continues, but at the expense of a large increase of steam to keep up the load.

Upon examination of the inside of the apparatus, buckets are found missing at one or several points of the single-disk turbine, and very often part of the disk itself is torn off. The explanation of the damage is that the water rushing back from the exhaust pipe (not necessarily from the condenser itself) follows the walls of the turbine casing, usually made so as to facilitate the outward flow of the exhaust steam, and comes into contact with the turbine disk whose peripheral velocity is about 1200 feet per second. The impact due to the velocity of the water alone would not affect the disk if the disk were standing still, but with the disk revolving at a high rate of speed the buckets cannot be expected to resist even a relatively small mass of water, no matter how strong they may be.

With multistage turbines there is some discrepancy of opinion as to the effect of a back flow of water. The turbine man is almost certain that the trouble was caused by water; the operator points out that the last wheel, which is directly in the path of the water, is either intact or else damaged, quite evidently, from the after effects of what took place in the adjoining stages. In these stages there seems to have been an explosion, buckets on the rotary and guide vanes in the diaphragm being blown on either side of a particular spot. Hence the hasty conclusion that the steam path was improperly designed, to offset which the turbine man points out that the machine had operated satisfactorily until the accident happened.

Close examination of the casing and moving parts in several wrecks of large machines has enabled the writer to offer a theory of the occurrence. In the first place the condensing process, even with the best types of condenser, is not absolutely continuous, and that for several reasons: the moisture of the exhaust steam is not a fixed quantity, it fluctuates even when the load is kept steady, due to initial conditions in the boiler, entrained water in steam lines, partial condensation or reëvaporation in the various passages of the turbine, etc.; the presence of variable quantities of air in the exhaust steam; the variable work of the air pump or of the circulating pump; the variable temperature of the water of circulation, etc. In addition to these unavoidable causes there comes the variation of the load itself, which sets the governor to work on the throttle valve, thereby causing a strong disturbance in the steam flow throughout the unit, and causing the single or double columns of steam in the exhaust line to oscillate.

The exhaust line is seldom lagged; hence the condensation begins as soon as the steam leaves the last rotor, and

the water agglomerates in the form of a sheet flowing against the wall of the exhaust duct. When this sheet of water is interfered with, through the oscillations of the exhaust steam, it is likely to move to and fro, pendulum like, and all the time acquiring more bulk, until a critical period arrives when it is thrown violently one way or the other. When thrown toward the turbine it follows the walls of the pipe, then those of the exhaust head, and if the latter are as shown in Fig. 2, taken from Stodola's work on steam turbines, the water following the direction of the arrows impinges against the buckets of the last rotor. The interbucket spaces are then sealed for a very short period, during which the exhaust flow is interrupted; the pressure builds up meanwhile in the several preceding stages, compressor fashion, until it becomes strong enough, with the help of the centrifugal action of the rotors proper, to overcome the inertia of the water seal. Then there is a sudden release, like a mild explosion, which produces intense vibrations throughout the fixed and moving vanes, and the latter give way, bending on either side of the explosion center, as observed.

The exhaust line is usually provided with traps or drain pipes capable of taking care of quite an accumulation of water, but yet unable to cope with sheets of water in rapid motion. Disturbances of this kind are likely to take place in any condensing installation, and as a remedy, the use of baffles located in the exhaust line has been advised at times. However, with the intelligent design of the casing reproduced in Fig. 1, the return water would follow the arrows and not come in contact with the rotor; and it should prove that the best safeguard is that which protects the turbine at the very danger point, hitherto so made as to invite danger.

ALBERT E. GUY.

Trenton, N. J.

Lubricator Condensing Chamber

Mr. Wallace in his letter under the above title in the July 4 number seems to be a little mixed up on his lubricator-condensing proposition. He wishes to know why the hollow chamber is in the top of the lubricator. What does it do? What is it for? Is it in the right place?

This chamber is not necessary to the working of the lubricator, but it is a part of it and is in the right place at the top of the lubricator. It is used to condense steam and is very essential as a reservoir, as the lubricator is never filled to the top.

The condenser will condense a greater volume of steam in a given time in its present place than it would 2 feet above the lubricator.

Mr. Wallace leads his readers to believe that this chamber is to condense steam continuously. The only advantage I can see in placing this chamber as he proposes would be to have the condenser 2 feet above the lubricator; this would fill the pipe below it very quickly and start the lubricator much sooner than if the chamber was attached direct to the lubricator. Then the chamber would have to be filled and afterward the pipe.

Mr. Wallace, like a great many engineers, seems to think that every time he fills the lubricator it is imperative that he should blow a great volume of steam through the drain valve. If such is the case, I will not censure Mr. Wallace for wanting to place a part of the lubricator at the highest point obtainable as it is quite worrisome to wait for steam to condense so you can start oil to your engine or pump cylinders.

If Mr. Wallace will close the water valve and open the drain, and then screw out the filler plug, the lubricator will empty very quickly.

I have seen lubricator condensers which had not been emptied in six months, but they should be blown out four times a year.

If Mr. Wallace will follow my suggestions he will not think it worth while to make the changes he has mentioned.

J. W. DICKSON.

Memphis, Tenn.

Some Flue Gas Analyses

Since the appearance in the June 6 number of POWER of the article on "The Value of Flue Gas Analysis," by Joseph W. Hays, we have completed our second installation of oil-firing apparatus for the Canadian Pacific Railway on two of its passenger steamers, and have made tests of gases that may be of interest to your readers, especially to those in the oil-burning sections of the country.

All the tests were made under ordinary working conditions while the ships were making normal high speed, and as there are no stack dampers in these vessels, the results show that highly efficient combustion can be maintained with proper equipment if intelligently handled.

In the case of the last installation on the steamship "Princess Charlotte," it is necessary to maintain an overload on its boilers of practically 100 per cent. in order to make schedule time. The vessel carries six Scotch boilers with 18 furnaces, having a total heating surface of 14,892 square feet, and the engines develop approximately 6000 indicated horsepower.

Both ships were converted from coal burning to oil, with a resultant saving in fuel and labor equivalent to 30 per cent. of the former coal consumption.

The accompanying tables show the results of some flue-gas analyses.

TABLE 1. RESULTS OF FLUE-GAS ANALYSES ON THE STEAMSHIP "PRINCESS MAY"

Date	CO_2	O	CO
April 6, 1911			
11:30 p. m.	14.6	0.0	0.14
11:45 p. m.	14.8	0.8	0 0
April 7			
12:20 a. m.	14.2	0.0	0.6
4:20 p. m.	14.8	0.2	0.2
4:30 p. m.	12.6	3.4	0.0
5:00 p. m.	14.2	1.0	0 0
5:20 p. m	14.6	0.8	0 0
April 8, arrived Skagway 2:30 p. m.; left for south 11:00 p. m.			
April 9			
1:45 a. m.	14.6	1.2	0.0
2:05 a. m.	14.6	1.6	0.0
2:18 a. m.	14.8	0.8	0.0
2:32 a. m.	14.4	1 2	0.0
2:43 a. m.	14.8	1.0	0 0
10:30 a. m.	14.8	1.0	0.0
5:50 p. m.	14.4	2.0	0.0
6:10 p. m.	14.0	2.8	0.0
6:20 p. m	14.8	1.8	0.0
11:10 p. m.	14.2	1.4	0.0
11:45 p. m.	13.0	3.0	0.0
April 10			
10:25 p. m.	14.8	1.2	0.0
11:00 p. m.	13.8	2.4	0.0
April 11			
8:15 a. m.	14.4	2.2	0.0
8:30 a m.	13.8	2.2	0.0
9:00 a. m.	13 0	3.0	0.0
9:40 a. m.	15.0	0.0	0.0
10:00 a. m.	14 8	1.0	0.0
Averages	14.3	1 4	0 004

TABLE 2. RESULTS OF FLUE-GAS ANALYSES ON THE STEAMSHIP "PRINCESS CHARLOTTE"

Date	CO_2	O	CO
June 6, 1911			
	14.2	1.8	0.0
	15.2	0.6	0.0
	14.0	2 2	0.0
	15.0	0 2	0.0
	15.1	1.0	0.0
Averages	14.7	1.16	0.0

J. F. BUMILLER,
Manager, National Fuel Oil Appliance Company.

Los Angeles, Cal.

Value of CO_2 Recorder

In the July 11 issue, in his reply to my criticism of his article on the value of the CO_2 recorder, Mr. Vassar makes the following points:

First, that I was in error in believing that the principal object in his writing the article was to question the usefulness of CO_2 recorders, and to put in doubt the honest purpose of those who are developing and endeavoring to introduce this apparatus. He calls particular attention to his statement that CO_2 recorders have a proper place in many boiler rooms; that what he intended to show was that "for the average boiler plant without a technical man to interpret results the CO_2 recorder is a rather questionable investment." In this I agree with him fully. In a boiler house in which the average intelligence of the operatives and foremen does not rise above the ability to shovel coal and watch the steam gage and water glasses, and where there is no technical supervision available, the installation of a CO_2 recorder would be an absolute waste of money. But, does not the same thing hold true of water meters and coal-weighing machines? Would it

mean any more to the average coal heaver to take an occasional look at the registers of the water meter and the coal weigher than to squint at the record on the CO_2 machine? Enough technical knowledge combined with practical training to interpret the readings is required in either case or beneficial results cannot be expected. On this point probably all concerned will agree.

The second point Mr. Vassar makes is that I neither advanced any argument tending to disprove his statement nor furnished experimental data to shed light on the problem. Mr. Vassar's statements are:

First. "Therefore, while high efficiency cannot be expected with low CO_2 it does not follow that high CO_2 is accompanied by high efficiency." Second. "I do not believe that CO_2 records are at all dependable as measures of furnace efficiency."

Mr. Vassar seems to labor under the misconception that the record of CO_2 in the flue gas, to be of any value, should be a measure of efficiency. CO_2 is not claimed to be a measure of boiler efficiency. The per cent. CO_2 is a practically correct measure of the excess of air, hence a measure of the weight of flue gas, and in combination with the temperature of the flue gas it is a practically correct measure of the sensible heat wasted up the chimney and, since the waste up the chimney constitutes from 60 to 90 per cent. of the total heat loss, including the loss due to unconsumed combustible gases and carbon in the ash, it follows that the per cent. of CO_2 in combination with the temperature of the escaping flue gases is not a direct measure of but a true index to boiler efficiency. The weight of coal burned and water evaporated are the most important factors in determining the heat utilized; while the percentage of CO_2 in the flue gases and the temperature at which the gases escape are the most important factors in determining the heat wasted.

A knowledge of both of these is desirable if not absolutely necessary to intelligent boiler-room management. Since CO_2 is not a direct measure of efficiency the diagrams as presented by Mr. Vassar prove nothing against the CO_2 recorder.

In his rejoinder to my criticism of his former article, Mr. Vassar devotes considerable space to the difficulty of sampling the gas, and says, "An automatic CO_2 recorder would indeed be a valuable adjunct to the boiler room if a fair sample could be secured with any degree of certainty." I admit that it is difficult if not indeed impossible to get a true average sample of the gas flowing at any moment; that the perforated pipe is impractical, and that the open-ended tube is by no means an ideal sampling arrangement. None the less, if the open end of the sampling tube is placed near the center of the gas current just before entering the uptake, continuous sampling results in a practically correct average of the total flow of gas.

CO_2 recorders require regular, intelligent attention. So does every operating device in the power plant from the steam turbine and dynamo, with all their appurtenances, down to the water pump. Few if any power-house appurtenances require less intelligent attention than a CO_2 recorder; many of them require a great deal more, and they get it, but the CO_2 recorder too frequently does not.

The reason is not far to seek: In the first place, no man's job depends on it. The man in charge of a pump knows that he must keep it in running order or lose his job, and he gives the pump the proper attention. Make a man's job depend on the regular operation of the CO_2 recorder and you will be surprised what little trouble it will give.

In the second place, the CO_2 recorder is looked upon by the boiler-house boss, in whose charge it is generally put, as an interloper, a despicable telltale. It does not help him keep up steam or reduce his labor. It exposes his shortcomings to the superintendent and as a rule he would rather see it out of commission than running, and if left to him it will not run.

In concluding his rejoinder Mr. Vassar asks me "to submit results of a series of commercial boiler tests disproving my (his) statement that CO_2 recorders at the present stage of the game are not trustworthy as measures of efficiency." Since CO_2 by itself it not claimed to be and in the nature of things cannot be a measure of efficiency, his request is not germane to the subject under discussion.

EDWARD A. UEHLING.

Passaic, N. J.

Cost of Furnace Upkeep

In an editorial in the July 4 issue some questions are asked concerning the "Cost of Furnace Upkeep."

Labor for laying linings and arches costs about $6 per eight-hour day, and it will take a mason at least 2½ days to lay them. In the linings every fifth or sixth course should be laid as headers. The number of firebrick necessary can best be computed by the engineer in charge as bricks vary in size in different localities. Then, too, the quality varies, but the engineer should insist upon using a brand that is satisfactory. Some kinds peel and flake off and others crack so badly that they cannot be used a second time. Firebrick costs from $22 to $30 per thousand. Fireclay costs $1 per 100 pounds. Salt added to fireclay mortar, in the proportion of 1 to 10 pounds of clay, helps it to set better. In laying the brick I have found it good practice to make the mortar very thin, dip the bricks into it and then lay them close. The mortar will allow for expansion and contraction. Walls laid with a thick mortar soon get shaky as the draft draws out the fine particles.

An arch made largely of iron, with the iron exposed to the fire, is worthless. The iron burns and wastes away and the work must be done over again. They last four months and cost about $10, plus $5 for labor and $5 for material, plus the time lost for the boiler.

A company in Wisconsin makes an arch lined with firebrick. The iron is not exposed, and the arch will last about 21 years at a cost of about $20 plus the charge of $10 for setting. The firebrick setting above the arch will need replacing at the end of one year at a cost of $18.

Boilers with overhanging fronts are cheaper to keep in repair, for then the arches need not be so wide, the space between the boiler sheet and the arch can be kept tight and the sheet at a head-end seam is amply protected from the fire.

Boilers with flush fronts must have the sheet at the seam covered. Some arches sag when they get hot and the seam becomes exposed to the fire. Then, too, the front edge of the arch burns off and lets the brickwark fall down.

Sometimes the arch tilts forward, but it can be jacketed back in place and new brick put under it. The mortar must be thin, or it will fall out and the arch will tip over.

If clinkers form on the side walls, care should be used in loosening them. If not, the brick will break out with the clinkers and soon the wall is undermined.

The back arch should be of the same material as the fire-door arches. A practise of using old grate bars, angle irons, etc., has crept in, but this means hot Sunday work for the firemen and additional expense to the firm.

I tried using cement with the fireclay. It cracked everywhere it should not and stuck so hard to the bricks that they had to be crushed to loosen it.

Boiler fronts with two doors using two arches are preferable to two doors touching and using one long arch. The long arch gives less service at a greater proportionate cost. When boilers are washed the walls and arches should be inspected. Often a pail of mortar and a few bricks applied at the critical time will save $10 in a repair bill, beside the loss of time for the boiler. Side walls should stand two years, but in many cases they must be relaid in six months.

ROY V. HOWARD.

Tacoma, Wash.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Distance Between Bearings

What is the safe distance apart to put line shaft bearings from 1 inch diameter of shaft and upward with and without pulleys between the bearings?

H. E. C.

Two formulas for determining the distance between the bearings of shaftings given in Kent are

$\sqrt[3]{873\,d^2} = L$ *for bare shafting*

and

$\sqrt[3]{175\,d^2} = L$ *for shafting carrying pulleys*

in which

L = Distance between bearings in feet.

d = Diameter of shaft in inches.

Standard Pipe Flange Dimensions

Is the table of flanged fittings in the June 27 issue now standard?

D. F. D.

The dimensions given in the table referred to are those recommended by the joint committees for the National Association of Manufacturers, Master Steam and Hot Water Fitters' Association and the American Society of Mechanical Engineers. Most of the large manufacturers are members of one or the other of these societies, and the dimensions given are quite sure to become standard.

Latent Heat in Steam

How many B.t.u. are there in the latent heat of steam?

A. J. S.

The latent heat of steam is found by subtracting the heat in the water from the heat in the steam. For instance: At atmospheric pressure the total heat in one pound of steam is 1150.4 B.t.u. above 32 degrees, and in a pound of water at the same temperature there are 180 heat units,

$$1150.4 - 180 = 970.4,$$

the number of latent heat units in a pound of steam at atmospheric pressure.

Condenser Cooling System

How many square feet of cooling surface are required in a surface condenser to condense one pound of steam per hour?

J. A. D.

With cooling water at 60 degrees, 1/13 of a square foot of cooling surface will condense one pound of steam per hour in ordinary condenser practice.

From 1½ to 3 square feet of cooling surface are allowed per horsepower, according to conditions.

Economy of Intermittent Speed

Is there anything saved by running a 14x36-inch Corliss engine four minutes at 125 revolutions per minute and then five minutes at 20 revolutions per minute, working that way automatically?

I. N. S.

So far as the engine itself is concerned, it will develop a given amount of power with less steam if it is run under constant conditions at its point of maximum efficiency than if run in the intermittent way described. There may be an advantage in running it that way, however, when the conditions and demands of the service to which it is devoted are taken into account.

Measurements for Snap Rings

How would you get the measure for and have made a set of snap rings for a solid piston?

S. P. R.

Snap rings are usually made of a thickness equal to 1/30 of the diameter of the cylinder plus ⅛ inch. The width must, of course, be an easy fit in the groove. They are generally turned to a diameter about ¼ inch larger than that of the cylinder before being cut for cylinders up to 20 inches in diameter, enough being cut out to allow them to spring into the cylinder. For larger diameters the rings are turned proportionately larger.

Hollow Staybolts

Why, when hollow screwed staybolts, having the hole not less than ½ inch in diameter and ends riveted over, are used, may the maximum allowable pitch be increased by the mean diameter of the staybolt?

H. S. B.

The supporting power of hollow staybolts, having a comparatively large bore, is found to be considerably greater than when the same amount of metal is concentrated in a solid bolt of smaller diameter. The threads are coarser, and have a greater hold, and there is a greater proportion of the sheet included in the span directly between the bolts.

Compression in Compound Engines

Is it easier to regulate the compression in a tandem than a cross-compound engine, and if so, why?

P. J. M.

In a tandem-compound engine the compression necessary to take up the momentum of the crosshead, piston and connecting rods can be divided between the two cylinders, and a comparatively small amount of compression on the large piston has a correspondingly greater effect.

Cylinder Arrangement in Compound Engines

Which is the better arrangement in a tandem-compound engine, to have the low-pressure cylinder nearest the crank, or *vice versa*?

C. A. E.

The arrangement is largely a matter of construction. The large cylinder is better adapted for direct bolting to the frame, and the small cylinder, especially if it overhangs, is better at the end. The low-pressure cylinder is cooler, and less heat will go from it by conduction to the frame and guides.

Position of Water Glass Washer

A claims that the proper place for the little brass washer in the water-glass connection in the water column is in the bottom of the stuffingbox. B claims that the proper place is on top of the gasket in the nut. Who is right, A or B?

M. J. M.

The office of the washer is to prevent the packing from being twisted or turned by the friction of the nut while being tightened, and should be placed between the packing and the nut.

Change of Water Level

Why does the water rise in a boiler after the engine is shut down?

W. B. L.

If there is any change in the apparent water level in the boiler when the engine is stopped it is because the pipe arrangement is such that there is less pressure upon the water under the steam outlet while the engine is running, causing the water to rise in that region when steam is being drawn off.

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POWER

Power Transmission on Oil Power Vessels

The fact that a double compromise in speed must be effected in applying the steam turbine to ship propulsion is an old story—the now familiar Melville-Macalpine gear owes its development to that condition of affairs, and several systems of electrical transmission have been devised with the object of relieving both the turbine and the propeller of the opposite speed handicaps to which they are ordinarily subjected.

With the advance in the application of internal combustion engines to marine service this question of power transmission from the engine to the propeller takes on increased importance, not so much because of speed requirements as on account of the lack of inherent flexibility which is characteristic of all types of internal combustion engine. A marine engine must, of course, be able to start instantly in either direction and go to full speed in a few seconds; in order to obtain this facility of operation the internal combustion engine requires substantial aid from some such auxiliary as compressed air or a clutch and reversing gear, the latter being impractical for equipments of several hundred horsepower.

The use of compressed air for starting gas engines naturally suggests its use as a medium of power transmission between the engine and the propeller, but even the most superficial consideration of that plan is sufficient to condemn it. Electrical transmission appears to be the only practical means of obtaining a flexible connection between the two extremes of the power plant, and its advantages are very great. Aside from the obvious features of independence of speeds at the engine and the propeller and the ability to start the propeller promptly in either direction and give it maximum acceleration, there is the enormous advantage that the propeller can be controlled entirely from the pilot house or bridge by means of a small lever such as is used to operate "telegraph" signals on large ships; no signals to the engineer are required and therefore there is no opportunity for misunderstanding or delayed response.

Of course, there are the extra weight and cost of the motive power equipment, which constitute serious disadvantages. They do not seem to be prohibitive in the case of large interurban railway cars, however, where almost exactly the same kind of problem exists. The use of gasolene-electric cars is increasing rapidly; why not crude-oil-electric boats?

A Chance for a Career

Under the above alluring title *The New York Times* expatiates editorially, with that degree of smug cocksureness so characteristic of the uninformed or inadequately informed, upon the alleged magnificent opportunities that hydroelectric development in this country is creating for technical graduates. The Feather River development of the Great Western Power Company is cited as an example and the *Times* asserts that "in the history of the country's industries there has never been so pronounced a demand for the services of technical experts." It fails, however, to shed any light on the important subject of remuneration; the editor contents himself with a glittering generality to the effect that "young men will hardly hesitate" in choosing between classical and technical education "if they wish to shape their careers toward *moneymaking*." The italics are ours.

Of course, by applying suitable definitions almost any statement can be made to withstand attack. If the editor of *The New York Times* considers receiving a salary of any amount between $1200 and $4000 a year as "moneymaking," his assertion is justifiable though his definition is—well, peculiar. We are familiar with the general facts relating to the "careers" of a good many technical and scientific men but we cannot think of a dozen who could, even by the most indulgent interpretation of the term, be considered "moneymakers." Practically all of the persons who have made money conspicuously from engineering undertakings have done so by commercial ability, not by the application of technical education.

However, technical education is priceless to the man who wants to earn his living by engineering, and there are even better uses of one's brains than moneymaking; the men who follow engineering because they love it usually get a good deal more out of life than do those who make use of it and them (the

engineers) to amass wealth. But that does not prevent us from feeling grimly amused at the idea of advising young men to take a technical education in order to become moneymakers.

The Old, Old Question

There was once a bicycle rider who tightened up the bearings of the wheels of his machine in order to retard his progress, but he was about to go down a steep incline. One could scarcely imagine anyone doing such a thing if he desired to make upward progress. Nevertheless, there are approximately a dozen proposed new license laws and proposed changes to existing license laws before the lawmakers in several States and cities in which engineers are asking the authorities to maintain a tax of $2 or more for each license issued and another tax for their renewal. These license laws are favored by some engineers because their aim is to protect life and property and raise the wages of the engineer. It is extremely doubtful if another body of men can be found in the whole country who ask to be taxed in order that the public may be protected from their ignorance and carelessness.

If these laws provide for the proper inspection and maintenance of boilers, every engineer and fireman is justified in working for the enactment of them; but there is no sane reason why the men should be taxed for the enforcement of the laws. The public should be safeguarded against danger from unsafe boilers, but it should pay for the protection, just as it pays for protection against fire risks, burglars and murderers.

Aside from the unjust imposition upon the licensed men of the cost of licensing them, the requirements for obtaining licenses need intelligent revision. Nine-tenths of the questions asked by an examiner have no direct relation to the safety of a boiler. How can questions dealing with condensers, valve-setting, heating, electric and hydraulic elevators, etc., affect the condition of a boiler or influence the manner in which it is to be managed by the applicant? The only value they have is in indicating that the man who can answer them correctly is intelligent enough to handle boilers in a safe manner. The possession of a license should be practically a proof of the ability of the holder to operate boilers and engines intelligently from the viewpoint of safety first, economy secondarily. The ability to generate a kilowatt-hour of energy at a low cost at the switchboard is no guarantee of ability to keep the boiler room in a safe condition. A good many power plants which their engineers strive to operate economically are operating with unsafe boilers.

Another bad feature of the license-law situation as it now exists is that after a man has obtained a license he can operate the plant of which he has charge with practically no interference or supervision on the part of the examiners. Once a year a State or city boiler inspector may come around and examine the boilers, if they are not insured, but he is ignorant as to how the plant is operated between visits.

Engineers should work for the passage of laws that will really protect the public and raise the standard of steam-plant operation but the expense of executing these laws should be borne by those who are benefited.

Master Mechanic or Chief Engineer

A master mechanic recently stated that it should be the duty of every competent man holding such a position to be familiar with the operation and construction of his power plant, so that, if so inclined, he can improve its efficiency.

This utterance shows that at least one master mechanic believes that if the efficiency of his steam plant is to be bettered it will be through his own efforts rather than those of the man in charge of the steam apparatus. It is reasonable to suppose that there are many other master mechanics who hold this view. One man who has a number of steam engines in his plant recently stated that although the men engaged in operating them were intelligent in a way they could not make detailed reports of power-plant operation and happenings in a clear, intelligent manner. These men would not be considered competent to maintain an efficient plant or to make alterations or additions to it.

If one can judge from the appearance of things, there are a number of engineers who are incapable of taking complete responsibility for the steam plants which they are operating. In any such case the master mechanic of the establishment, if he is interested in the economical operation of the plant, as he should be, will naturally assume the position of chief engineer himself and the steam man will be restricted to a subordinate rank. When a change like this occurs, the operating engineer, being incapable of handling the steam branch of the business, should not resent the "intrusion." On the other hand, if the steam engineer is thoroughly competent, the master mechanic will find that there are objections to his interfering with the management of the steam plant.

There are many master mechanics who are also the chief engineers of their plants, and it is probable that most of them would be only too glad to put the steam plant in the hands of trustworthy men and relieve themselves of all responsibility, if they could find such men. A competent chief engineer must of necessity have the executive ability to handle the men under him and also the mechanical training necessary to enable him to manage the mechanical equipment. This does not confine his ability to seeing that the engines and boilers are put in service at certain times, but includes the supervision of every detail that can in any way influence the cost of developing power. It is more important for him to know when to make a repair than it is to know how to do so after an accident has happened. It is necessary also to look ahead and plan for future improvements and additions which make for economy.

When an engineer can manage a steam plant in a practical and efficient manner he need have no fear of interference from the master mechanic.

The Human Element

In estimating the cost of power production one important factor is too often neglected. Interest, depreciation, taxes and coal per kilowatt-hour are calculated to the third or fourth decimal place and the prevailing market rates for attendance are always considered but the human element is usually either ignored or forgotten. Consequently, the actual performance of the plant seldom equals the estimated results.

Realizing this, many managers have tried the experiment of interesting the men in the economical operation of the plant. This is brought about by some system which makes the employee's income increase with the employer's gain. Under such conditions the employees feel that they are a part of the institution and have an interest in the work with which they are identified.

Given fair treatment and recognition for service actually performed, most men will soon drop those personal characteristics which make for friction, because it will pay to do so.

In the earlier days of steam engineering the piston was made more or less steam tight in the cylinder by means of a packing of braided hemp which was renewed as the discretion or caprice of the operator dictated. The idea that the piston packing should be a part of the machine furnished by the builder was never entertained until an itinerant machinist designed a form of metallic packing which he made on the premises whenever he effected a sale. If it is a builder's duty, as it seems to be, to furnish pistons and valves which will not permit the leakage of steam, why is it not equally his duty to provide for waste around valve stems and piston rods by furnishing a steam-tight, easy-running metallic packing which shall be comparatively durable?

Heating and Ventilation

Considered as power plant problems. Layout and operation of systems and apparatus

Proposed Basis for Rating House-Heating Boilers and Furnaces*

By Frank L. Busey

A study of several hundred tests made by the University of Illinois engineering-experiment station on various types and sizes of small boilers and furnaces, and also of tests made on other and larger units, the results of which could be relied upon, are the data upon which the results given in this paper were based.

The practice of using 10 or 12 square feet of heating surface in a boiler as equivalent to 1 horsepower is hardly warranted in house-heating boiler work, since the rate of combustion, ratio of heating surface to grate area and consequent variation in efficiency may invalidate such a system. As a matter of fact, the amount of surface required may vary anywhere from 5 to 50 square feet per horsepower.

Various forms and combinations were tried and discarded, and it was finally decided to use the term horsepower as the unit of heating effect, as applicable to both steam boilers and warm-air heaters. Taking the American Society of Mechanical Engineers' standard of 34.5 pounds of water evaporated per hour from and at 212 degrees Fahrenheit as equivalent to 1 boiler horsepower, gives

$$34.5 \times 970.4 = 33{,}479 \text{ } B.t.u.$$

as also equal to 1 horsepower. In the case of the boilers, the equivalent evaporation from and at 212 degrees Fahrenheit per hour divided by 34.5 gives the horsepower developed. With the warm-air furnaces the pounds of air heated per hour multiplied by the rise in temperature and the specific heat of air, gives the number of B.t.u. delivered per hour to the air. This in turn divided by 33,479 gives the horsepower developed by the furnaces.

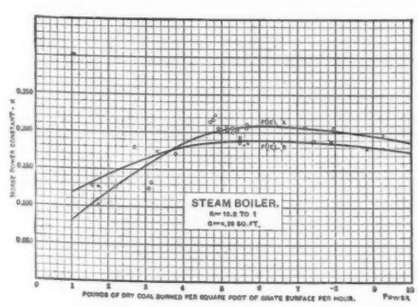


Fig. 1. Relation of Horsepower Constant to Rate of Combustion

Having determined the total horsepower developed by the various units tested, the next step was to find some dimension, relation or ratio to which the horsepower developed could be referred, and so obtain a unit to which the horsepower would bear some definite relation. It was found that the horsepower developed upon each square foot of grate by 1 pound of dry fuel burned per hour on each square foot of grate surface was remarkably constant. This is true for any one boiler or furnace when burning dry coal per square foot of grate per hour between the limits of 4 to 8 pounds for small units and 5 to 10 pounds for the larger units, when using coals of a similar quality. This is shown clearly in Figs. 1 and 2 for two separate units, each using two classes of fuel. This value, termed a horsepower constant and designated by K, was obtained for each test analyzed by dividing the horsepower developed by the grate area, and this quantity in turn by the rate of combustion. Since frequent reference will be made to the grate area and the rate of combustion for any one heating unit, these terms will be designated by G and F, respectively.

While K was constant for any one unit and for a similar quality of coal, yet different units gave different values of K. It was found that the higher values of K were obtained from units having a greater number of square feet of heating surface for each square foot of grate, this number being the ratio of the total square feet of heating surface to the square feet of grate. This ratio will be designated by R. For a similar quality of coal, the values of K from the different units were then plotted to the corresponding values of R, as shown in Fig. 3. It is seen that there exists a very definite relation between the values of K and R.

This relation appears reasonable, for, other conditions being equal, higher efficiencies will be obtained with higher values of R. Thus in practice it is customary to use additional sections, resulting in higher values of R, and also greater capacity. The relation between K and R was found to be independent of the size of the unit tested, inasmuch as increase in size is usually accompanied by an increasing value of R. This accounts for the higher efficiencies obtained from large units.

It is well known that the use of various grades of coal results in different efficiencies in the same unit. From a study of the tests made it was found that the coals used could be divided into three general classes, designated as classes A, B and C. Class A includes anthracite, coke and semi-bituminous coals. Class B includes good grades of bituminous coal, such as is mined in Williamson, Franklin and Saline counties of the southern district of Illinois. Class C includes the poorer grades of bituminous coals. It has already been pointed out that when using any one class of coal a definite relation exists between K and R. Although different values of K are obtained when using the different classes of coal, yet each class has its own definite relation between K and R. This is well illustrated in Fig. 3, where the three curves refer to the different classes of fuel.

Table 4 shows the average proximate analyses of the three classes of coals during these tests. The term "good" or "poor," as here used, refers to the performance of the coal when used in small house-heating units and not for large power boilers. The comparatively low furnace temperatures attained in the smaller furnaces make it impossible to

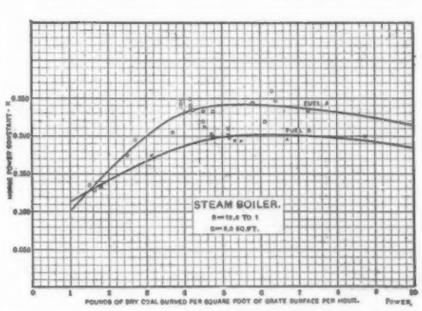


Fig. 2. Relation of Horsepower Constant to Rate of Combustion

*Abstract of a paper read before the American Society of Heating and Ventilating Engineers, Chicago, July 6 to 8.

burn completely the excessive amounts of volatile matter in these coals designated as "poor bituminous." For this reason they soot up the flues and cut down the efficiency.

It is customary, to get the same heating effect, to build warm-air furnaces with a much greater value of R than is used with steam and hot-water boilers. This is because more heat can be transmitted through 1 square foot of heating surface to water than to air. Tests show that in warm-air furnaces R must be from two to three times as large as in boilers to give the same efficiency, and corresponds very closely with practice.

As a result, K does not bear the same relation to R in furnaces as in boilers, due to the greater value of R necessary in furnaces to obtain the same value of K. It follows that the relation of K to R varies not only with the different class of fuels used but with the different type of heating unit (see Fig. 3).

Fig. 3 shows the relation between the horsepower constant K and the ratio of total heating surface to grate area R. The general relation of the curves A, B and C for furnaces and curves B and C for boilers is very similar, the better fuels giving the higher efficiencies. This difference increases slightly as the value of R increases. An increase in the value of R is accompanied by an increase in the value of K only in so far as the value of R is confined to the limits found in practice. These limiting values of R for boilers in which this relation was found to hold are between 8 and 15. This range for furnaces, however, is from 15 to 38. It is quite probable that the higher value of 15 set for R in the case of boilers can be increased to 20 or even above, but it was impossible to obtain sufficient data to prove absolutely this point. Nevertheless the indications are that this is the case. There would eventually come a point where the temperature of the flue gases would be reduced to so nearly the temperature of the surrounding water or air that no further gain would be effected by increasing the radiating surface.

Curve A, in the case of the boilers, shows a much greater difference in favor of the high-carbon over the bituminous coals as the radiating surface is increased. This difference is probably due to the comparatively restricted flue passages as ordinarily used in steam and hot-water boilers. In warm-air heaters the radiator is usually made of such ample proportions that less difficulty is experienced with soot deposits. If the ratio of heating surface to grate could be increased, and at the same time the gas passages kept of ample size, curves B and C would no doubt tend to assume an inclination more nearly like that of curve A.

It is evident that an increase in the radiating surface of boilers is productive of greater gain in efficiency when using Class A fuels rather than the ordinary bituminous coals. With boilers having a low value of R a good grade of bituminous coal may be expected to give as good results as with anthracite or eastern semi-bituminous coals.

As the values plotted in Fig. 3 can be represented by straight lines within the limits of the tests used, the next step is to derive a formula and determine the proper constants applicable to the various sets of conditions.

Let

G = Area of grate surface in square feet;

F = Pounds of dry fuel burned per square foot of grate surface per hour;

R = Ratio of total heating surface to grate area;

K = Horsepower constant = horsepower developed per square foot of grate, per pound of dry fuel burned per square foot of grate per hour;

$h.p.$ = Total horsepower developed;

C_1 and C_2 = Constants for any one set of conditions.

It will be noticed that the average deviation of the points from the lines as drawn is less than 3 per cent., and that the maximum deviation is 7.5 per cent.

TABLE 2. VALUES OF HORSEPOWER CONSTANT K FOR VARIOUS VALUES OF RATIO R, STEAM BOILERS

R Ratio of Total Heating Surface to Grate Area	Fuel A			Fuel B			Fuel C		
	K Horsepower Constant	B.t.u. per Hour Delivered to the Water	Square Feet of Radiation Served	K Horsepower Constant	B.t.u. per Hour Delivered to the Water	Square Feet of Radiation Served	K Horsepower Constant	B.t.u. per Hour Delivered to the Water	Square Feet of Radiation Served
8 to 1	0.166	5,560	22.2	0.177	5,925	23.7	0.159	5,320	21.3
9 to 1	0.181	6,060	24.2	0.182	6,090	24.4	0.163	5,455	21.8
10 to 1	0.196	6,560	26.2	0.187	6,260	25.0	0.167	5,590	22.4
11 to 1	0.211	7,065	28.3	0.192	6,430	25.7	0.171	5,730	22.9
12 to 1	0.226	7,565	30.3	0.197	6,595	26.4	0.175	5,865	23.5
13 to 1	0.241	8,065	32.3	0.203	6,785	27.1	0.179	6,000	24.0
14 to 1	0.256	8,570	34.3	0.208	6,960	27.8	0.183	6,135	24.5
15 to 1	0.271	9,070	36.3	0.213	7,130	28.5	0.188	6,280	25.1
16 to 1	0.286	9,575	38.3	0.218	7,300	29.2	0.192	6,420	25.7
17 to 1	0.301	10,075	40.3	0.223	7,465	29.9	0.196	6,550	26.2
18 to 1	0.316	10,575	42.3	0.229	7,665	30.7	0.200	6,690	26.8
19 to 1	0.331	11,080	44.3	0.234	7,835	31.3	0.204	6,830	27.3
20 to 1	0.346	11,580	46.3	0.239	8,000	32.0	0.208	6,965	27.9

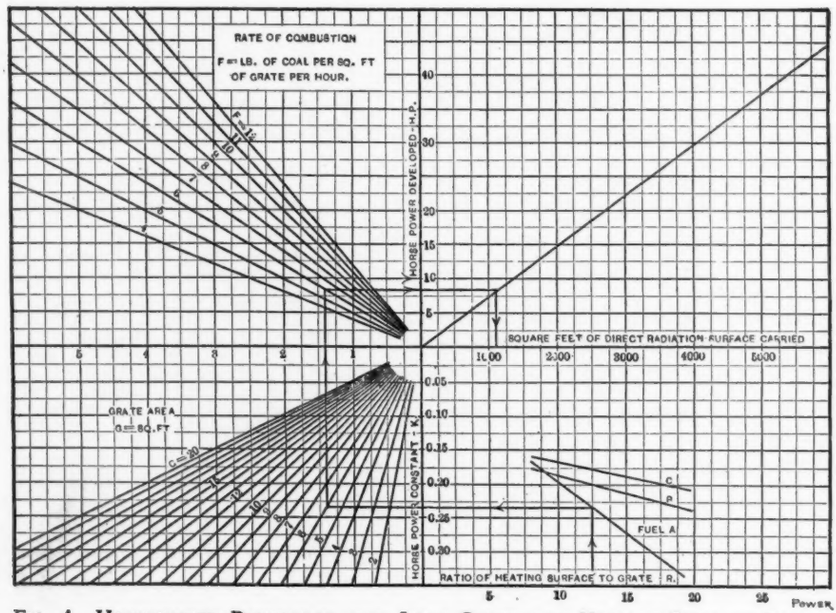

RATE OF COMBUSTION
F = LB. OF COAL PER SQ. FT OF GRATE PER HOUR.
HORSE POWER DEVELOPED - H.P.
SQUARE FEET OF DIRECT RADIATION SURFACE CARRIED
GRATE AREA G = SQ. FT.
HORSE POWER CONSTANT - K
FUEL A
RATIO OF HEATING SURFACE TO GRATE - R.
Power.

FIG. 4. HORSEPOWER DEVELOPED AND LOAD CARRIED BY VARIOUS SIZES OF BOILER

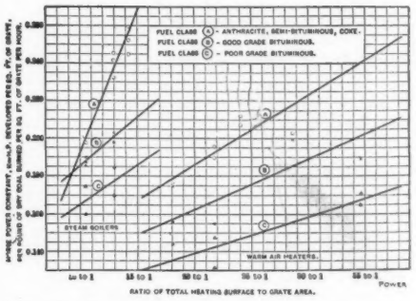

FUEL CLASS (A) - ANTHRACITE, SEMI-BITUMINOUS, COKE.
FUEL CLASS (B) - GOOD GRADE BITUMINOUS.
FUEL CLASS (C) - POOR GRADE BITUMINOUS.
STEAM BOILERS.
WARM AIR HEATERS.
RATIO OF TOTAL HEATING SURFACE TO GRATE AREA.
Power.

FIG. 3. RELATION OF HORSEPOWER CONSTANT TO RATIO OF HEATING SURFACE TO GRATE AREA

This shows how closely actual performance, as shown by a study of several hundred tests, may be approximated by the use of the formulas and constants given in Table 1.

Then these lines are represented by the formula

$$K = C_1 R + C_2 \qquad (1)$$

The American Society of Mechanical Engineers' standard of 34.5 pounds of water evaporated as 1 horsepower was used; hence one horsepower = 33,479 B.t.u. This in turn divided by 250 gives the number of square feet of direct radiating surface carried--taking the standard of 250 B.t.u. emitted per square foot per hour. With warm-air furnaces the B.t.u. delivered to the air per hour divided by 1.4356 (the heat required to raise one cubic foot of air from zero to 70 degrees Fahrenheit) gives the equivalent cubic feet of air heated from 0 to 70 degrees Fahrenheit.

TABLE 3. VALUES OF HORSEPOWER CONSTANT K FOR VARIOUS VALUES OF RATIO R, WARM AIR HEATERS

R Ratio of Total Heating Surface to Grate Area	Fuel A			Fuel B			Fuel C		
	K Horsepower Constant	B.t.u. per Hour Delivered to the Air	Equivalent Cu.Ft. of Air Heated per Hour from 0 to 70°F.	K Horsepower Constant	B.t.u. per Hour Delivered to the Air	Equivalent Cu.Ft. of Air Heated per Hour from 0 to 70° F.	K Horsepower Constant	B.t.u. per Hour Delivered to the Air	Equivalent Cu.Ft. of Air Heated per Hour from 0 to 70° F.
15 to 1	0.168	5610	3906	0.151	5040	3511	0.131	4385	3054
16 to 1	0.171	5730	3993	0.153	5130	3573	0.133	4455	3103
17 to 1	0.175	5855	4078	0.156	5220	3636	0.135	4520	3148
18 to 1	0.179	5980	4165	0.159	5310	3699	0.137	4585	3194
19 to 1	0.182	6105	4251	0.161	5400	3761	0.139	4655	3243
20 to 1	0.186	6225	4338	0.164	5490	3824	0.141	4720	3288
21 to 1	0.190	6350	4424	0.167	5580	3887	0.143	4785	3333
22 to 1	0.193	6475	4510	0.169	5670	3950	0.145	4855	3382
23 to 1	0.197	6600	4597	0.172	5760	4012	0.147	4920	3427
24 to 1	0.201	6725	4683	0.175	5850	4075	0.149	4990	3476
25 to 1	0.205	6845	4769	0.178	5940	4138	0.151	5055	3521
26 to 1	0.208	6970	4855	0.180	6035	4204	0.153	5120	3566
27 to 1	0.212	7095	4942	0.183	6125	4266	0.155	5190	3615
28 to 1	0.216	7220	5028	0.186	6215	4329	0.157	5255	3660
29 to 1	0.219	7340	5114	0.188	6305	4392	0.159	5325	3709
30 to 1	0.223	7465	5201	0.191	6395	4455	0.161	5390	3754
31 to 1	0.227	7590	5287	0.194	6485	4517	0.163	5455	3800
32 to 1	0.230	7715	5373	0.196	6575	4580	0.165	5525	3849
33 to 1	0.234	7835	5459	0.199	6665	4643	0.167	5590	3894
34 to 1	0.238	7960	5545	0.202	6755	4705	0.169	5660	3943
35 to 1	0.242	8085	5632	0.205	6845	4768	0.171	5725	3988
36 to 1	0.246	8210	5718	0.208	6935	4831	0.173	5795	4034

In this formula C_1 and C_2 have definite values for any one type of heating unit in various sizes when using one class of coal.

The various sets of values for C_1 and C_2 have been carefully determined and are tabulated in Table 1. Thus the formula for warm-air heaters for a fuel in class A becomes

$$K = 0.0037\ R + 0.112$$

For any unit under consideration this formula gives us the value of K, or the horsepower that is developed on each square foot of grate surface by each pound of fuel burned per square foot of grate per hour. Knowing K, the grate area, and the most desirable rate of combustion, the total horsepower developed can be readily determined by the formula

$$h.p. = K \times G \times F \qquad (2)$$

TABLE 1. VALUES OF C_1 AND C_2 FOR THE FORMULA $K = C_1 R + C_2$

	Fuel	C_1	C_2
Steam Boilers	Class A, anthracite, semi-bituminous, coke	0.0150	0.046
	Class B, good grade bituminous	0.0052	0.135
	Class C, poor grade bituminous	0.0041	0.126
Warm-air Heaters	Class A, anthracite, semi-bituminous, coke	0.0037	0.112
	Class B, good grade bituminous	0.002	0.110
	Class C, poor grade bituminous	0.0020	0.101

Referring to Fig. 4, assume a boiler with a value of $R = 12.5$, a grate area of 6 square feet and a rate of combustion of $F = 6$ pounds. Then, if anthracite is to be used, follow the vertical line upward from the point where $R = 12.5$ until it intersects the line A. From this intersection follow the horizontal line to the left through the point $K = 0.235$ until it intersects the line $G = 6$. From this point trace the vertical line upward until the line $F = 6$ is reached. Following the horizontal line to the right from this point it is seen that under these conditions the horsepower developed is 8.5. On tracing it farther to the right until the diagonal line is intersected, then dropping downward on the vertical line, the equivalent heating surface served is found to be 1150 square feet.

In the same manner for warm-air furnaces on Fig. 5, the values can be traced as shown. From $R = 25$, fuel B, $G = 7$, $F = 6$, to 7.8 horsepower or a heating effect of 180,000 cubic feet of air warmed per hour from 0 to 70 degrees Fahrenheit. This operation can be reversed, with either Fig. 4 or 5, thus working from the amount of heat needed back to the unit to be selected.

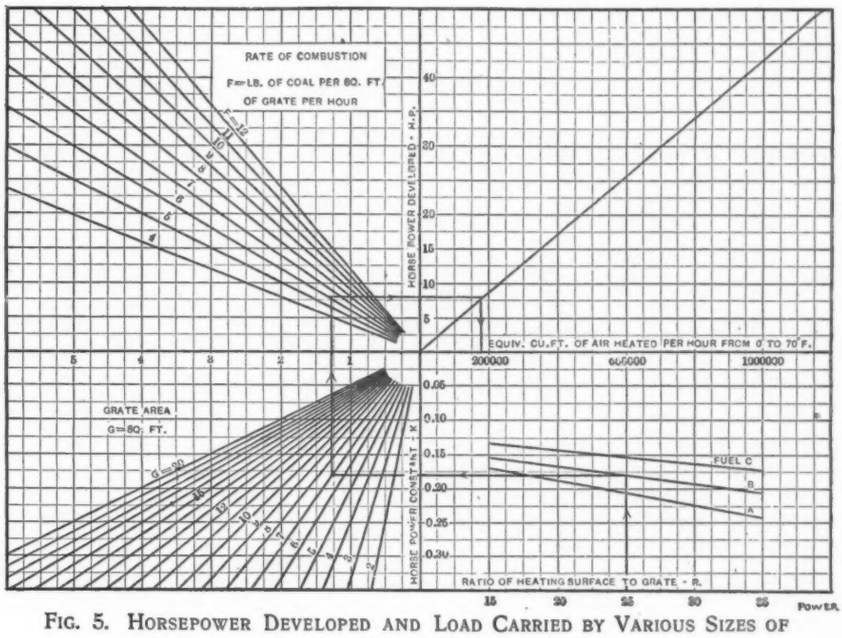

RATE OF COMBUSTION
F=LB. OF COAL PER SQ. FT. OF GRATE PER HOUR
HORSE POWER DEVELOPED - H.P.
EQUIV. CU.FT. OF AIR HEATED PER HOUR FROM 0 TO 70° F.
200000
600000
1000000
GRATE AREA
G=SQ. FT.
HORSE POWER CONSTANT - K
FUEL C
RATIO OF HEATING SURFACE TO GRATE - R.
POWER

FIG. 5. HORSEPOWER DEVELOPED AND LOAD CARRIED BY VARIOUS SIZES OF FURNACES

Any basis of rating selected for heating installations should be equally ap-

plicable to steam and hot-water boilers and to warm-air furnaces, as they are used for similar requirements.

A unit of rating should be of such a character that it may be easily expressed and comprehended, of some standard form that may be compared to that used for power-boiler rating.

The term horsepower as used in the American Society of Mechanical Engineers' rating for power boilers is a measure of heat delivered, is applicable to both boilers and furnaces and fulfils better than any other term the above requirements.

TABLE 4. AVERAGE PROXIMATE ANALYSIS OF FUELS USED, FUEL AS FIRED

	Fixed Carbon, Per Cent.	Volatile, Per Cent.	Moisture, Per Cent.	Ash, Per Cent.	Sulphur, Per Cent.	B.t.u. per Lb.
Class A—Anthracite	78.25	7.13	3.49	11.15	1.30	12,820
Coke	80.50	3.26	6.30	9.94	0.90	12,015
Semi-bituminous	74.57	18.79	1.49	5.15	0.74	14,780
Class B—bituminous	49.07	34.91	6.84	9.19	1.82	12,250
Class C—bituminous	41.59	38.05	10.79	9.57	2.97	11,340

The heat generated, or the power developed, on each square foot of grate by the combustion of 1 pound of fuel on that square foot of grate has been found to be practically a constant, under the conditions as already outlined. This is true of both boilers and furnaces.

This constant bears a definite relation to the size of the heating unit as expressed by the ratio of its total heating surface to grate area. Based on the results of actual tests, this relation can be expressed by a simple formula and the above constant determined for different units and sets of conditions.

The difference between the horsepower developed by actual test and that calculated by the use of this formula is comparatively slight. The calculated horsepower may be used in connection with the ordinary heating job with entire satisfaction.

The use of the diagrams, Figs. 4 and 5, will expedite the work of calculation, and for any ordinary case is sufficiently accurate. Where greater accuracy is desired the formulas (1) and (2) and the values of C_1 and C_2 from Table 1 may be used.

Sulzer Brothers, of Winterthur, Switzerland, have taken out a patent for a governor for steam engines, more particularly steam turbines, which is operated by the pressure induced in a working fluid (as oil) by a small centrifugal or impeller pump, run directly from the turbine shaft. The oil is fed to the pump at constant head, an overflow being provided to insure this constancy, and the pressure, depending directly upon the velocity of the direct-connected impeller, is made to control the valve which governs the admission of steam to the turbine.

CORRESPONDENCE

Heating System Improvements

The equipment of a certain vacuum-heating system consists, in part, of an open feed-water heater. As indicated in Fig. 1, a drain pipe from the main exhaust line had been run to the heater and connected to the opening, that was originally designed for a vent, directly to the atmosphere. This left the heater without adequate relief for the air liberated from the feed water and therefore the air continued to accumulate to the exclusion of a proper volume of exhaust steam. This arrangement was, of itself, sufficient to keep the heater from doing its best work. To make matters worse, the apparatus was further handicapped by a direct overflow connection to the same sewer that received the discharge from a pneumatic sewage ejector; the overflow valve was never tightly closed on account of a single float doing the double duty of controlling the inflow as well as the overflow, the two valves being connected through a system of levers.

The unimpeded overflow passage thus arranged simply afforded a convenient channel through which the air already lodged in the heater by natural entrainment was supplemented with a goodly portion of the blast of air blown through under relatively high pressure at each ejectment. As a consequence, the heater frequently failed to heat the water, the temperature of which was often as low as 60 or 65 degrees—practically the temperature of the supply main.

The performance of this heater was one of the first things noted by a chief engineer newly arrived on the job, and the alterations he made to bring it up to a proper standard of efficiency were quite simple, but decidedly effective. These changes are shown in Fig. 2, and comprise an air vent and a loop inserted in the overflow pipe to form a water seal.

This work was done in the summer time, when an ample volume of exhaust steam was always available for heating the feed water and the vacuum-heating system was out of service. The advantage to the apparatus was immediately apparent; for, while the daily average of feed-water temperature theretofore had been about 155 degrees, which average, however, was based upon a series of individual readings covering a wide range, it promptly went up to approximately 208 degrees Fahrenheit.

Regarding the proposition from the theoretical standpoint of 1 per cent. saving of fuel for each eleven degrees gain in the temperature of the feed water, the temperature increase indicated above should have shown a saving of 4.82 per cent. As a matter of fact, the daily coal consumption showed an actual per cent. reduction very close to this value, the calculation on both sides being reduced to a uniform basis of heat energy consumed per unit weight of water evaporated.

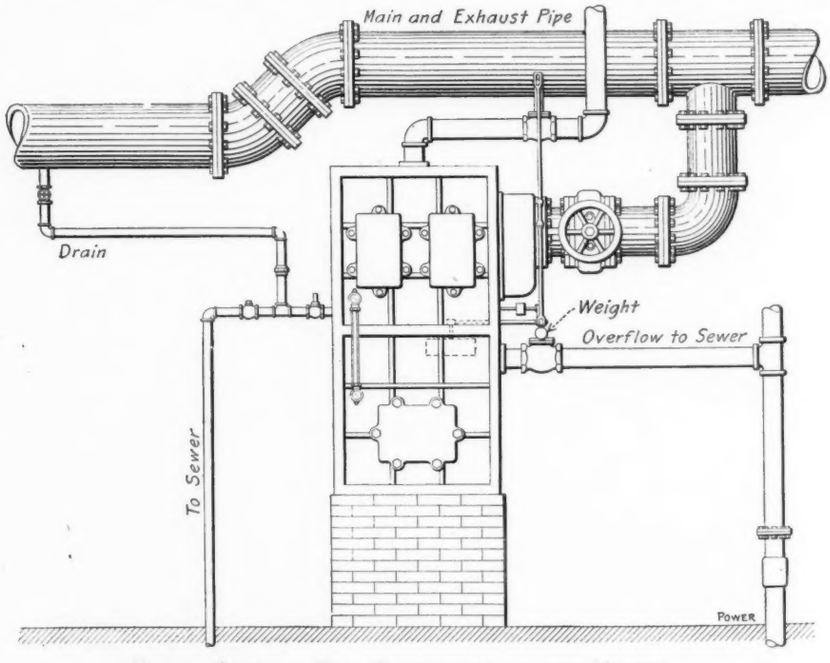


FIG. 1. ORIGINAL PIPE CONNECTION TO THE HEATER

Another very excellent device adopted, and one which also had a direct influence in establishing a practically uniform heater temperature at the highest point, but at a time when the entire volume of exhaust steam was utilized in the heating system, was the substitution of needle valves for the ordinary ½-inch gate valves previously used to control the flow of injection water to the vacuum pumps, and the installation of thermometers in the discharge pipes from these pumps.

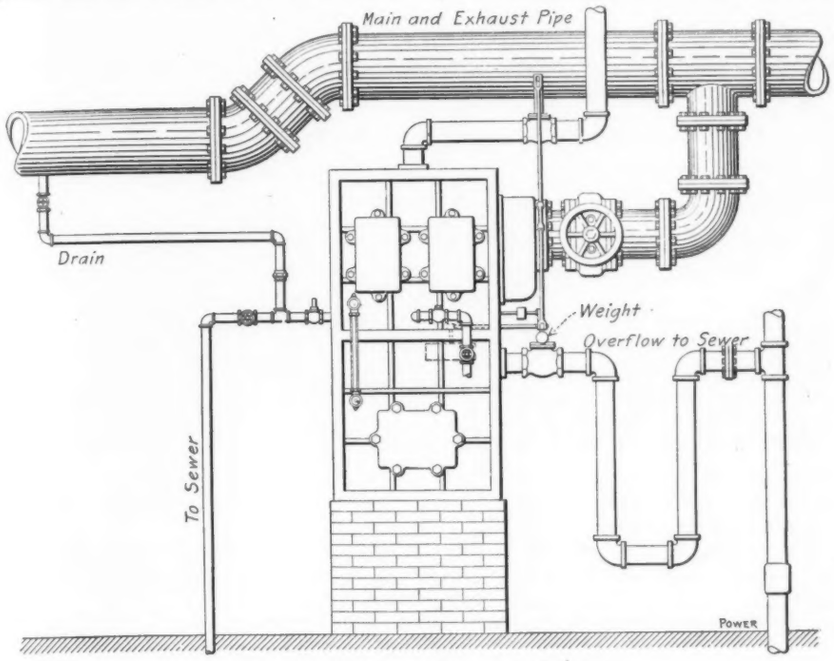


FIG. 2. SHOWING CHANGED PIPING

This improvement was of particular value to the engineer on watch as he could intelligently adjust the quantity of injection water to varying conditions, the discharge from the pumps being maintained constantly at 190 degrees, thus conserving the heat of the returns as well as economizing the water consumption. With the old arrangement, the only recourse was to open the ½-inch gate valve a few turns and trust to luck for the rest.

One other adjunct which the new chief deemed necessary, and one which greatly benefited the plant, consisted of a float-controlled automatic valve for admitting the make-up water. The outfit was already equipped with the customary overhead air-separating tank for receiving the discharge from the vacuum pumps, but the above mentioned accessory had been omitted, and an ordinary globe valve, manipulated by the fireman, had been used instead.

As the regulation of this valve solely depended upon the fireman, it is easy to understand that there were times when considerable heat energy, as well as good city water, was going to waste through the overflow to the sewer.

A. J. DIXON.

Chicago, Ill.

Piping Hot-water Heaters

I have found the usual practice in hot-water heating for baths, etc., to be that of running the hot water, or highest water connection of the heater, to the bottom or side of the reservoir, and the cold-water connection on the heater is made in a similar manner. This does not appear to be the best plan for good service.

Hot water is lighter than cold water, and, therefore, tends to rise to the highest point of the reservoir. If it mixes with cold water as it rises, it requires a long time before one is able to get any hot water at the faucet; that is, all the water above the level of the pipe must be heated to using temperature before the system is of service.

The hot-water connection should be made to the highest point of the reservoir and the cold-water connection to the lowest. When connected in this way, with the supply to baths, etc., taken from a high point on the reservoir, or from a tee connecting the reservoir, hot-water connection and supply to baths, etc. (which is the better arrangement), one may get hot water in a few minutes after the fire is started. Whatever hot water there is will be in the top of the tank and may be drawn direct by the user; and it is soon supplemented with hot water from the heater.

The supply water from the system should enter the bottom of the reservoir or heater, preferably into a tee in the cold-water connection.

I have had the hot-water connections fill with lime or scale, and consider this dangerous as I know of two heaters that burst from this cause, one wrecking the room above it. It also pays in quick service, and in a saving of fuel to keep the heaters clean.

ROY V. HOWARD.

Tacoma, Wash.

Engine Wrecked at Morgantown, W. Va.

On Wednesday evening, August 2, at the Sabraton works of the American Sheet and Tin Plate Company, Morgantown, W. Va., the engineer was almost instantly killed by the piston rod breaking in the crosshead at the eye through which the key passes to hold it in the crosshead. The engine, which was a Corliss 36x60 inch and made 30 revolutions per minute, was used to drive five hot mills. The piston rod, which was 6½ inches in diameter, knocked out the cylinder head and the upper left corner of the cylinder, the break plainly showing it had been cracked for some time. The engine was installed in a concrete pit about 5 feet deep, beside an exact duplicate. The engineer had just come on duty, had gone over his engines and was sitting down back of them and to one side when the accident occurred, throwing him a distance of 20 feet. From the effects of the scalding and bruising he received, the engineer died on the way to the hospital.

Educational Program of Modern Science Club

On September 26, from 7 to 10 p.m., a beefsteak dinner will be held in the grill rooms of the Modern Science Club, 125 South Elliot place, Brooklyn. An exposition and discussion of the proposed winter courses of study will follow.

In addition to the regular Tuesday night lectures the proposed educational program for the season of 1911-12 will consist of two courses of 30 lectures, each embracing subjects bearing directly on the vocation of steam-engine operation.

Beginning Friday, October 6, the first course will treat of mathematics for engineers, such as fractions, decimals, percentage, square and cube root, equations, the slide rule, natural philosophy and graphical statics.

On Monday, October 9, the second course begins. It will consist of lectures on fuels, their origin, nature, characteristics and analysis; steam, its generation and distribution in engines, pumps and turbines, theory and practice in the use of condensers, including their design and installation; steam-engine design and operation; theory of the turbine, both impulse and reaction; the indicator and its application; the slide-valve diagram and its value in steam-engine design; internal-combustion engines, including gas producers; flue-gas analysis and general engine- and boiler-room chemistry.

New Power House Equipment

What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news

Richardson Sight Feed Mechanical Lubricator

This lubricator is designed to supply oil to a cylinder just as it is required. The oil is fed drop by drop from the feed nozzles at any rate desired and is broken up into a number of small particles which are mixed with the steam entering the cylinder.

Each lubricator contains an oil reservoir, and a gage glass which shows the oil level. The pump body is made from a solid block of iron drilled out to receive the interior mechanism. The pumping plungers are made of polished-steel drill rod, working in die-molded babbitt bushings. The plungers are actuated by a common gear shaft that is operated from a driving lever which is connected to any convenient reciprocating part on the engine.

The operating mechanism runs in oil, almost entirely eliminating wear, and there are no projecting screws, nuts, ratchets, glasses under pressure or by-pass devices.

The operation of this lubricator will be made clear by referring to the accompanying diagram, in which the travel of oil from the reservoir to each individual feed line is shown. The oil is withdrawn from the reservoir through foot valve *A*, Fig. 1, by means of the plunger *B*, delivering the oil through the circulating channel *C*, into which are connected the adjustable needle valves *M M*, etc., for regulating the flow of oil through each drip nozzle *R*, Fig. 2.

All surplus oil not passed through the feed valves returns to the reservoir through the overflow ball check *D*, Fig. 1. The slight gravity head on each drip nozzle is constant, and the oil flows drop by drop at any desired rate into the chamber *G*, Fig. 3. The plungers for each individual feed line are actuated by the driving lever through a common gear shaft *H*. These plungers move up and down with every revolution of the engine and at each downward stroke chop off a small particle of the oil protruding from the chamber *G* and force it out through the check valves *K K* and into the feed lines *L*. The feed lines are always full of oil up to the check valve *N*, located at the point of delivery; thus every time the pump plungers descend they force a small particle of oil into the line, and release a corresponding amount at the terminal check valve. As the oil is forced through two check valves *K K* and a terminal check valve *N*, there is no liability of steam pressure backing up into the lubricator. The check valves are of a special steel-ball type, and are held in place on a bronze seat by a coil spring under a hexagon nut. The babbitt bushings *F* can be quickly taken up for wear by removing the hexagon cap nut *W* and slightly tightening up the valve cage *U*.

An important feature is the increased feed valve *P* located in the oil-circulating channel *C*. This valve is normally left open, but it can be closed should there be a temporary demand for increasing the amount of oil fed by shutting off the return to the reservoir and causing all of the oil delivered by the circulating plunger to flow through the feed nozzles, thus temporarily forcing the flow of all feeds. When the increased demand has been met, the increased feed valve *P* is opened. Thus the adjustment of the needle valves *M* need never be changed to meet a temporary demand for additional oil.

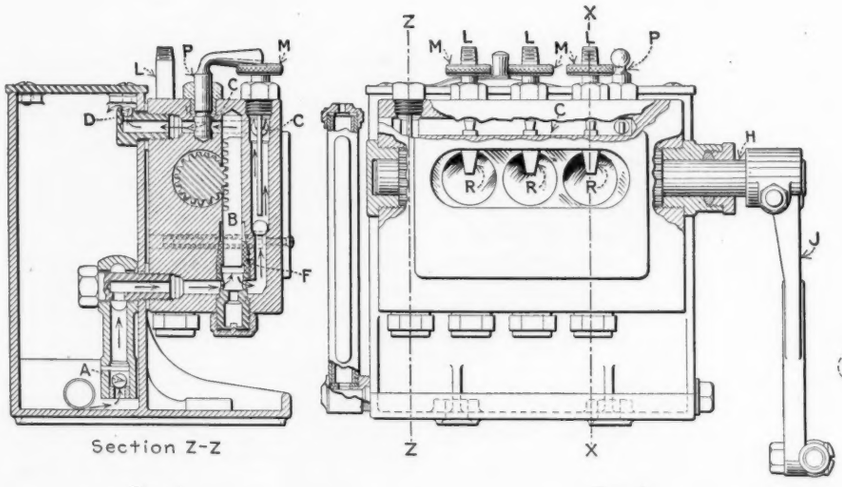


FIG. 1. FIG. 2.

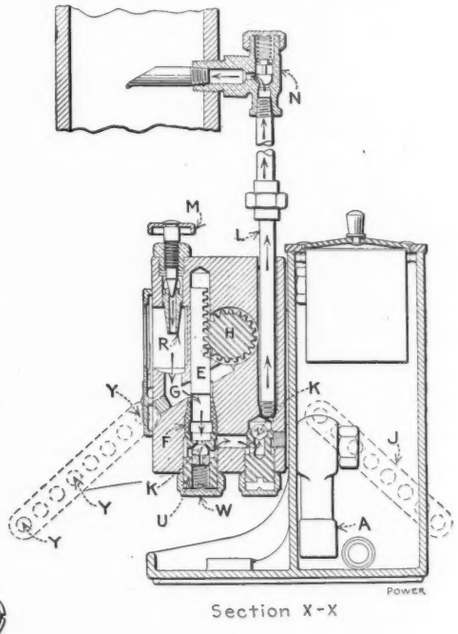


FIG. 3.

THREE VIEWS OF THE MODEL "M" MECHANICAL LUBRICATOR

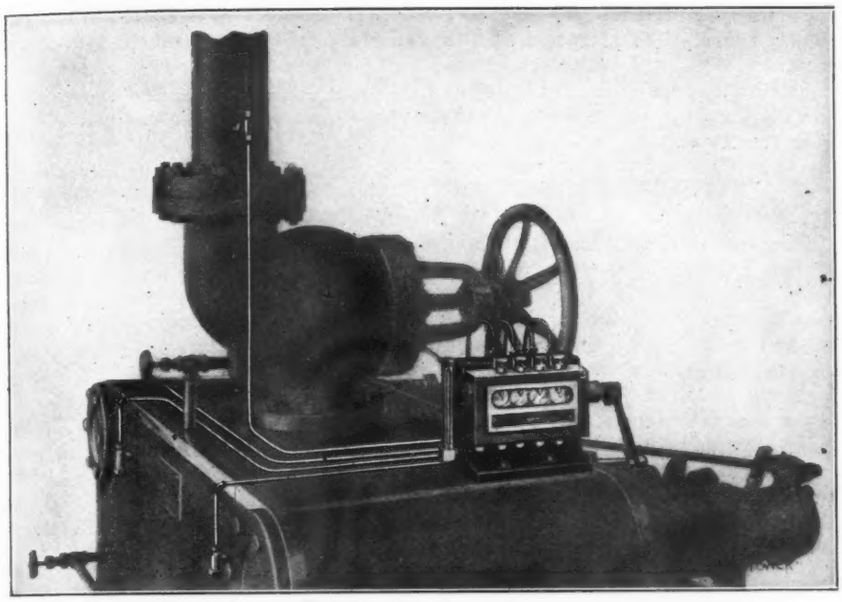


FIG. 4. A FOUR-FEED LUBRICATOR, SHOWING PIPE CONNECTIONS TO ENGINE

The driving lever *J* requires a reciprocating movement of from 72 to 96 degrees and can be moved into any position on the shaft. The necessary travel of the lever *J* can be adjusted by attaching the driving-rod pin in any one of the eight wristpin holes *Y*, thus permitting any straight-across travel from 3 7/16 to 8½ inches.

The glass protecting the feed nozzles is not under pressure, being held in position by a spring-retained brass frame which can be easily tipped forward for cleaning the nozzles.

This lubricator can be applied to any vertical, horizontal, simple or compound steam engine, and to all types of gas engines, ammonia compressors, high-duty pumping engines, air compressors, pumps or any machinery where it is desired to feed a predetermined amount of oil against pressure.

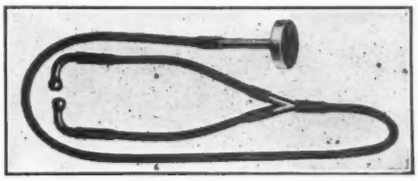
In Fig. 4 is shown the application of a four-feed lubricator, one oil pipe connecting to the steam pipe, two to the steam valves, and one pipe is led to the piston-rod metallic packing.

A pump having 22 feeds is shown in Fig. 5. It is used for automatically lubricating the bearings of high-duty elevator pumps or wherever a number of bearings are to be lubricated. These pumps are manufactured by the Richardson-Phenix Company, Milwaukee, Wis.

Hopewell Vibracator

A device for detecting knocks in an engine is shown in the accompanying illustration.

The head of the vibracator consists of a hardened-steel corrugated diaphragm that is made as thin as it is possible to draw the metal. The vibrations of the machinery are transmitted to the diaphragm which sets up vibrations in the air inclosed in the head and the amount of the air inclosed in the head is small, giving only necessary clearance for vibrations. These vibrations are transmitted through the air up the tubes into the ear tips and thence to the ear. The ear tips are held in position by springs inclosed within the rubber tubes. The diaphragm head is insulated on to the rest of the body by nonabsorbent material so that the vibrations are transmitted directly to the air.



HOPEWELL VIBRACATOR

The device is manufactured by Hopewell Brothers, Newton, Mass.

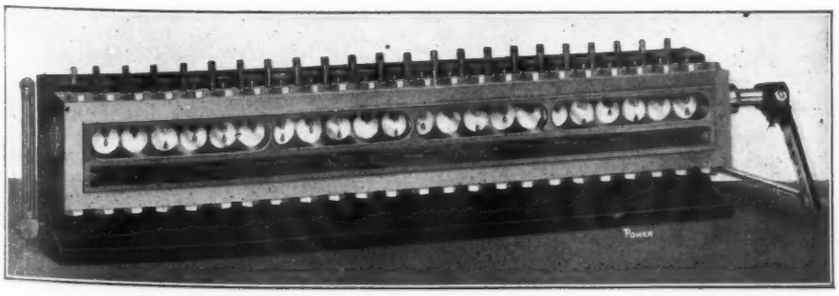
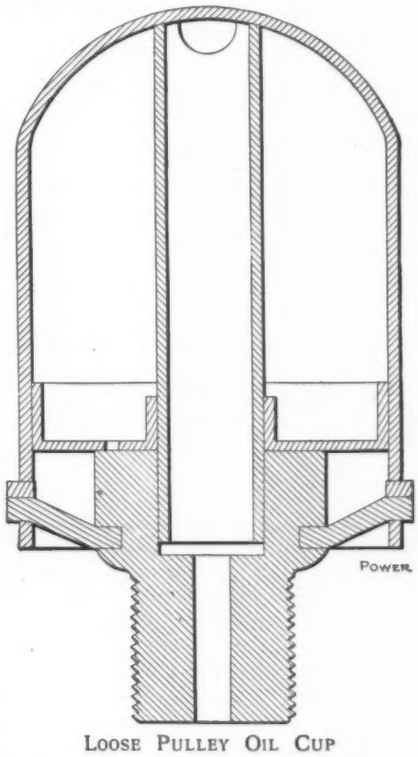


FIG. 5. A TWENTY-TWO FEED FORCE PUMP

Loose Pulley Oil Cup

The accompanying sectional view illustrates an automatic loose-pulley oil cup which will run from one to three weeks when filled once, according to the number of starts and stops, speed, etc.

As the pulley is brought into operation the centrifugal force throws the oil to the top of the cup and fills the feeding tube. When the pulley starts the next time a portion of the oil in the tube is fed to the bearing and the tube again fills with oil.



LOOSE PULLEY OIL CUP

The cup can be detached by hand from the nipple and can be removed, filled and replaced with the pulley in any position.

The cup is made of thin pressed steel by the American Specialty Company, Chicago, Ill.

To drive the machinery in a table and furniture factory in Schoharie, N. Y., two waterwheels have been installed on a small creek. In addition, the proprietor not only lights his house and factory, but a neighboring church and the main village street. This is but one of many instances where the high cost of fuel has caused the small user of power to utilize the natural advantages near at hand.

PERSONAL

G. H. Gleason has become vice-president of the Dexter Engineering Company, Incorporated, Providence, R. I. He was formerly engineering salesman of the Dodge Manufacturing Company, Boston, Mass.

Prof. Carl C. Thomas, of the University of Wisconsin, is now in Europe conducting some interesting investigations along engineering lines. It is his intention to remain on the continent until the summer of 1912. All correspondence should be addressed in care of Knauth, Nachod & Kühne, Leipsic, Germany.

Frank J. Wood, chief engineer for Marx & Rawolle, at Brooklyn, N. Y., is the first recipient of the gold medal awarded by the American Institute of Chemical Engineers for the best paper presented to the society. Such medals are to be awarded every three years, and Mr. Wood earned the first by the presentation at the Philadelphia meeting in 1909 of a paper upon "Glycerin Refining in Multiple Effect Stills." Mr. Wood is a prominent member of the N. A. S. E. and of the Modern Science Club, of Brooklyn, of which he was president for a number of terms.

Alonzo Pawling, president and treasurer of the Pawling & Harnischfeger Company, Milwaukee, and for many years a prominent figure in the machinery field, has disposed of his interests in the company and retired from business life.

In December, 1884, Mr. Pawling, together with Henry Harnischfeger, founded the Pawling & Harnischfeger Company, under the firm name of Pawling & Harnischfeger. From a small beginning the concern has grown to large proportions.

Mr. Harnischfeger becomes president and treasurer of the company; W. H. Hassenplug, vice-president; F. P. Breck, second vice-president, and S. H. Squier, secretary. There will be no changes in the policy of the company or in its organization.

Edwin M. Herr, who was elected president of the Westinghouse Electric and Manufacturing Company at a meeting of the board of directors held in New York, August 1, has been the first vice-president of the company and in charge of operation at East Pittsburg since June 1, 1905.

He was born in Lancaster, Penn., May 3, 1860, and upon completion of a common-school course, became a telegraph operator on the Kansas & Pacific railroad, with which company he remained for two years, being promoted to the position of station agent.

In 1881 he entered the Sheffield Scientific School of Yale, graduating with the class of 1884, and worked as an apprentice in the shops of the Pennsylvania Railroad Company at Altoona, Penn., during the two summer vacations. From 1884 to 1885 he was an apprentice at the West Milwaukee shops of the Chicago, Milwaukee & St. Paul Railroad, and then went to the Chicago, Burlington & Quincy Railroad Company as a draftsman in the mechanical engineer's office, later becoming assistant engineer of tests, and finally engineer of tests. From 1887 to 1889 he was superintendent of telegraphy, and from 1889 to 1890 was division superintendent of this road.

From 1890 to 1892 Mr. Herr was division master mechanic of the Chicago, Milwaukee & St. Paul Railroad at West Milwaukee which he left to accept the position of superintendent of the Grant

EDWIN M. HERR

Locomotive Works at Chicago. After remaining in this capacity for two years he became superintendent of motive power and machinery of the Chicago & Northwestern Railroad, and from June 1, 1897, to September 10, 1898, held the same position with the Northern Pacific Railroad.

In 1898 he became assistant general manager of the Westinghouse Air Brake Company and the following year was promoted to the position of general manager, which position he held until June 1, 1905, when he was elected first vice-president of the Westinghouse Electric and Manufacturing Company.

The other officers elected by the board of directors of the Westinghouse Electric and Manufacturing Company to serve with Mr. Herr were: Chairman of the board of directors, Robert Mather; vice-presidents, Loyall A. Osborne, Charles A. Terry, Harry P. Davis; acting vice-presidents, Henry D. Shute, George P. Hebard; comptroller and secretary, James C. Bennett; treasurer, T. W. Siemon; auditor, F. E. Craig.

SOCIETY NOTES

A special train will take the New England and New York delegates and vistors to the convention of the National Association of Stationary Engineers at Cincinnati. This train will leave Boston on Sunday, September 10, about 10 a.m. and another leave New York over the West Shore Railroad at 2:30 p.m. The Boston train will stop at Worcester and Springfield; the New York train at Highlands, Newburg and Kingston. The two trains will be joined at Albany and from that point proceed to Cincinnati, stopping at Syracuse, Rochester and Buffalo. Reservations and further information about the train may be obtained from James R. Coe, chairman of the transportation committee, 21 Maiden Lane, New York City.

BOOKS RECEIVED

PUMPING MACHINERY. By Arthur M. Greene. John Wiley & Sons, New York. Cloth; 703 pages, 5½x9 inches; 504 illustrations; indexed. Price, $4.

Experiments on the preservation of Westphalian coal under exclusion of air have been carried out by Dobbelstein, and are described in *Glück Auf!* of May 6. The coal consisted of lumps about ¾ inch in size and vessels of about ½ gallon capacity were charged with this coal and provided with air-tight covers. The vessels were further filled, either with water or with carbon dioxide, flue gases or sulphurous acid vapors. Samples were taken and analyzed after two weeks, three weeks and six months; at the end of this period no further weathering seemed to occur. Parallel tests were made with coal exposed to the air. The different sorts of coal did not all behave equally. But on the whole it would appear that little is gained by keeping the coal under water or in special atmospheres. The loss of gas was little affected by the treatment, and the difference observed might have been less noticeable if the coal had been taken as soon as mined, and not after intervals of from five to eight days; the weathering is most marked in the first days. If anything special is to be done for the storage of coal, roofs should be provided for the coal piles, and thermometers be inserted so as to obtain warning of any tendency to spontaneous combustion.—*The Engineer.*

A boiler explosion at one of the Pittsburg mills of the Crucible Steel Company of America on August 2, injured three men and damaged property to the extent of $10,000.

Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power

ARTHUR BRISBANE, whose editorials in the Hearst papers probably reach more people each day than those of any other man, and who gets $50,000 per year for writing them, recently addressed the Advertising Men's League of New York City on "Does Advertising Increase the Cost of Living?"

A few of his good remarks follow:

"The cost of living is a thing we cannot nail down. Years ago there was the same kind of talk about it. One of our early presidents said that the country was going to the dogs because you could not get a man for less than a hundred and fifty dollars a year."

"If there is any problem of cost of living it's because there has not been enough good advertising."

"In the first place I should say that advertising is to business and the community what speech is to the individual socially."

"Now, before man could talk, if he knew a thing, it was impossible to tell another human being about it."

"Imagine the waste of time, money and effort if the five million people in this community could not speak."

"One might know where such and such a thing might be had and could not tell the others, and each one would have to find it out for himself—where it was or where the best was. It would be an absolutely foolish waste of time if they could speak."

"Advertising is business speech."

"There is nothing about advertising except economy, except saving, except reduction of the cost of living."

"Of course, when you talk about advertising, manufacturing or agriculture you don't talk about stupid advertising, manufacturing or agriculture. Farmers that lose money don't count. Advertisers that try to persuade the public to lose money don't count."

"Advertising, of course, is a medium of distribution."

"Take this question about the lack of advertising. Suppose everyone stopped advertising, what would happen? Of course, you would have to talk, you would have to tell people things and very badly."

"Advertising does another thing which should be emphasized. *It makes the advertiser give hostages to commercial honor.*"

"If the manufacturer said 'this value has built these enormous factories, they have cost me millions and there is no insurance on them,' nothing would make him set fire to them. Now, the manufacturer's reputation is uninsured. He spends thousands in building something more solid than a brick building, and when he has built that up, built up that name, you cannot imagine that he would jeopardize that fortune by making his goods inferior."

A manufacturer has to live with and on the reputation of his goods, keep them sold and make them sell others.

That's why only reliable products can be continuously advertised.

That's why buying advertised products is the cheapest quality-insurance you can possibly get.

BUSINESS ITEMS

The Henry Vogt Machine Company, Louisville, Ky., reports recent orders for 11 Vogt water-tube boilers from 130- to 300-horsepower units, a total of 2420 horsepower.

Among substation equipment orders recently received by the Westinghouse Electric and Manufacturing Company is one from the Gainesville Railway and Power Company, Gainesville, Ga., for two 125-kilovolt-ampere, oil-insulated, self-cooling, 11,000-volt transformers; one switchboard and two regulators; constituting a complete substation equipment.

The Vilter Manufacturing Company, 854 Clinton street, Milwaukee, Wis., reports recent sales of 10 Corliss engines ranging in size from 9x24 to 24x42, including a 14x24x36 cross compound for the Aluminum Goods Manufacturing Company, Calumet, Mich. It has also received a number of recent orders for refrigerating machines from 5 to 100 tons capacity.

The location of the western office of the J. S. Bretz Company, of New York, has been changed to 504 Ford building, Detroit, Mich., where J. W. Hertzler, its western representative, will make his headquarters. A full sample line of F. & S. imported ball bearings, German steel balls, Star ball retainers, U. & H. master magnetos, Bowden wire mechanism, Hartford universal joints and clutches, and drop forgings will be displayed there for the convenience of the western trade.

NEW EQUIPMENT

Dorris, Cal., will install waterworks system.

Prairieburg, Iowa, will install a waterworks system.

Greenfield, Mass., will extend its water system.

Burlington, Ore., will install an electric-lighting system.

Tarentum, Penn., will establish a municipal waterworks system.

Cummington, Mass., contemplates increasing its water supply.

Vallejo, Cal., contemplates the erection of a municipal lighting plant.

The Bridgeton (N. J. Water Company will build a new pumping plant.

The Smith Paper Company, Lee, Mass., will install new power equipment.

Hastings, Mich., voted $120,000 bonds for municipal hydro-electric plant.

An engine house will be built for the Reading Railroad at St. Clair, Penn.

The Columbus (Ohio) Ice Company will make improvements to its plant.

The Russell & Erwin Company, New Britain, Conn., will build a pump house.

The Mayo Woolen Mill Company, Millbury, Mass., will install a new engine.

The Porterville (Cal.) Northeastern Railroad will build a new power plant.

Hemet, Cal., is planning for the installation of a waterworks and pumping plant.

Etna, Penn., has let contract for the construction of a new pumping station.

The Bradley Polytechnic Institute, Peoria, Ill., will erect a $25,000 power house.

Bonds for $3000 have been voted for a waterworks system at Courtland, Minn.

The Northern Cold Storage Company, Duluth, Minn., will erect a $43,000 addition.

The Steubenville (Ohio) Ice Company will erect an addition, doubling cpacity of plant.

An addition is being erected to the boiler house of Pratt, Read & Co., Deep River, Conn.

The Evans Syndicate will build a new street railway and power plant at Saskatoon, Sask.

The Worcester Suburban Electric Company will erect a power plant at Millbury, Mass.

The plant of the St. Albans (Vt.) Cold Storage Company was burned. Loss, $30,000.

The power plant of the Pressed Steel Wheel Company, Pittsburg, Penn., was destroyed by fire.

C. C. Swindell has been granted franchise to install an electric-light plant at Tenaha, Tex.

North Adams, Mass., has under consideration a municipal gas and electric-lighting plant.

The city of Bakersfield, Cal., plans to install new waterworks system at East Bakersfield.

The citizens of Pocahontas, Iowa, will vote on issuing $11,000 bonds for an electric-light system.

The plant of the Crystal Ice Company, at Pierre, S. D., was destroyed by fire. Loss, $20,000.

The city of Quebec, Canada, will spend $750,000 on waterworks extensions and new equipment.

The Eagle Ice and Coal Company, Philadelphia, Penn., will erect a new plant on Fifty-Sixth street.

Wilkie, Sask., will equip a complete new pumping plant and also electric-light and power plant.

Kamloops, B. C., is planning the erection of a 5000-horsepower power plant on the Barrier river.

Eugene, Ore., has voted $57,000 bonds for improvements and extensions to the municipal electric plant.

The Salisbury (Md.) Ice Company will rebuild its ice plant recently burned. J. D. Price, manager.

The Desert Power and Water Company, Kingman, Ariz., will increase the capacity of its power plant.

An electric-light plant will be installed at Glen Ullin, N. D. C. E. V. Draper, Mandan, N. D., engineer.

The Ohio Valley Electric Railway Company will erect an addition to its power house at Kenova, W. Va.

Jackson, Ga., will vote on issuance of $5000 bonds for extending its electric-light and waterworks.

The boiler room of the Wisconsin Iron and Wire Works, Milwaukee, Wis., was destroyed by fire. Loss, $15,000.

Gladstone, Ore., will install a new pumping plant and waterworks system. L. C. Kelsey, Portland, Ore., engineer.

The St. Gothard Gold Mining Company, Grass Valley, Cal., is planning the erection of an electric power plant.

Six turbine-driven generators will be purchased by the Birmingham (Ala.) Water, Light and Power Company.

Pasadena, Cal., is having plans prepared by A. L. Sonderegger, Los Angeles, for a modern waterworks system.

The C., R. I. & P. RR. will erect a new power house at Wentworth avenue and Forty-Seventh street, Chicago, Ill.

The Commonwealth Edison Company will build a $23,000 substation at 3431-33 North Whipple street, Chicago, Ill.

Ten thousand dollars has been appropriated at Webster City, Iowa, for the municipal electric-light and power plant.

The Pacific Power and Light Company, Portland, Ore., contemplates installing an electric-lighting system at Adams, Ore.

The city council, Palatka, Fla., has granted franchise to C. S. Hammatt to construct and operate an electric-light plant.

The St. Louis (Mo.) Independent Packing Company has awarded contract for the erection of a refrigerator building.

Seattle, Wash., is planning to increase the capacity of its power plant at Cedar Lake from 12,000 to 30,000 kilowatts.

The Cudahy Packing Company is making additions to its cold-storage plant at Fourteenth and Jones streets, Omaha, Neb.

HELP WANTED

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

WANTED—Chief engineer for a small power plant; one having experience with a.c. and d.c. machines. "E. P. E.," POWER.

WANTED—Thoroughly competent steam specialty salesman; one that can sell high-grade goods. Address "M. M. Co.," POWER.

AN ENGINEER in each town to sell the best rocking grate for steam boilers. Write Martin Grate Co., 281 Dearborn Street, Chicago.

AGENTS WANTED for first-class steam specialty in use throughout United States. Address C. S. Wood, 410 S. 15th St., Philadelphia, Penn.

WANTED—First class steam engineer with at least ten years' experience, to operate a turbine and reciprocating station having high and low tension distribution. Box 477, POWER.

THE VULCAN SOOT CLEANER offers an exceptional opportunity for power specialty salesmen. For further particulars address G. L. Simonds & Co., 802 Steinway Bldg., Chicago, Ill.

WANTED—An engineer in each city as agent for a high class water-back Scotch boiler, the most economical steam generator known to the trade. Kingsford Foundry & Machine Works, Oswego, N. Y.

Power plant equipment salesman wanted; one familiar with the line, to call upon jobbers, consulting engineers, and do missionary work among consumers; in your reply, state experience, references and salary wanted. Address Box 476, care POWER.

AGENTS WANTED—To sell lubricating oils, belts, hose, paint and varnish, for power machinery, to factories, mills, auto owners, stores, threshers; exclusive territory to right party; oil experience unnecessary. Manufacturers Oil & Grease Co., Cleveland, Ohio.

WANTED—By an engineering school in Canada, a technical graduate to assist in mechanical laboratory work and do some lecturing on mechanical subjects; salary $1200 for eight months' service; state nature of college course and engineering experience. Box 473, POWER.

WANTED—Erecting engineer; young man, unmarried, must have had practical experience with compressed air machinery; general knowledge of steam, gas engine and electrical machinery; must be able to superintend installation; will have to travel; state experience and references. Box 474, POWER.

SALESMEN WANTED—We are looking for three first class salesmen, engineers preferred, who can get the business and install packing when necessary; we do not object to booze, but do not want boozers; we will pay moderate salary and expense, until worth is demonstrated; Eastern and Middle States open; product now successfully in use, and duplicate orders coming in from the largest concerns in the country. Box 472, POWER.

SITUATIONS WANTED

Advertisements under this head are inserted for 25 cents per line. About six words make a line.

POSITION WANTED by steam engineer as chief in plant up to 500 horsepower, or as second engineer in larger plant; first class license; seven years' experience; three years electric; own indicator; 24 years old, married; strictly sober; A-1 references. P. O. Box 320, New Comerstown, Ohio.

MECHANICAL ENGINEER, 34, American, married, 10 years' experience, Latin-American countries, designing, construction, testing and operating gas power plants, open for engagement as superintendent of light or power plant, mining or milling work, business ability, can represent machinery house, correspondence in Spanish, French and German; prefer foreign countries, Latin-American or Hawaiian Islands. Box 475, POWER.

KEYSTONE GREASE

KEYSTONE TRADE MARK REGISTERED U.S. PAT. OFF.

"Help Yourself—They're Free"

Can Of Keystone Grease, an Engineer's Cap and a Brass Grease Cup

And don't delay; take advantage of this offer *today*.

For remember that by helping yourself to the Cap, Cup and Can of Keystone Grease you are helping to make your work in the engine room easier and more productive.

Because to use even one can of Keystone Grease is to discover the sure way of making your fuel last longer and your engine and machinery run more smoothly.

For Keystone Grease is the wasteless lubricant; it stays on the bearings, every bit of it being used up in useful work.

It is a scientific production containing nothing but high-grade refined petroleum oils which never become rancid or decomposed.

It requires no friction heat to feed.

Remember:—This coupon brings the Free Can of Keystone Grease, the Brass Grease Cup and the Engineer's Cap at once.

KEYSTONE LUBRICATING CO.

Executive Offices and Works

21st, Clearfield and Lippincott Streets **Philadelphia, Pa.**

Branch offices and warehouses in the principal cities of the United States.
Agencies in the principal countries throughout the world.

No Connection With The Oil Trust

A SAMPLE FREE

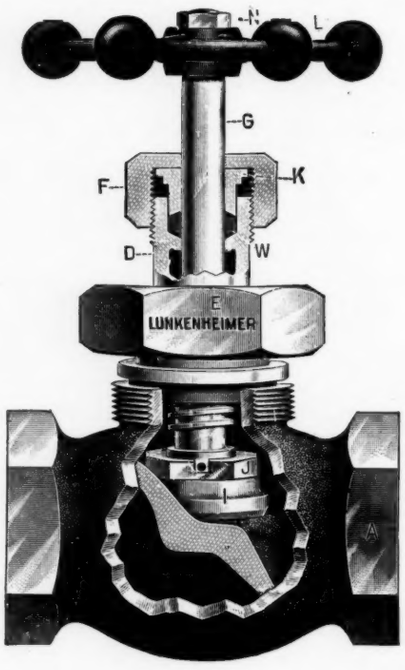
KEYSTONE LUBRICATING COMPANY, Phila.

Please send us a large size sample of Keystone Grease, sufficient for testing purposes, and one Brass Grease Cup and Free Engineer's Cap. It is thoroughly understood that no charges whatsoever will be made for sample, cup or cap, and all express charges will be paid by you.

Name of bearing where sample will be tested . . . . . .
Its width . . . . . Combined horse power of engines . . . . . .
Tap size of cup . . . . . .
FIRM. . . . . .
Per . . . . . .
City . . . . . State . . . . . .
Size cap required . . . . . **CUT OUT THIS COUPON AND MAIL IT TO-DAY** Dept. B—8-22-11

Just A Few Minutes

Spent In Regrinding These Valves Makes Them As Good As New



Lunkenheimer Regrinding Valves

Require no extra parts when the seating faces are worn. They can be reground quite a number of times with very little trouble and without necessitating their removal from the connecting pipe.

NOTICE the manner to which the hub is secured to the body. This union connection permits the valve to be quickly taken apart for regrinding or repair and also protects the threads from the corroding action of the steam.

The valves can be packed under pressure when wide open and the areas throughout the bodies are largely in excess of the nominal diameters of the connecting pipes.

The material used is a bronze composition containing a high percentage of copper and tin and the workmanship is of the highest grade.

Lunkenheimer Globe, Angle and Cross Regrinding Valves are made in two styles, one with inside screw and the other with outside screw and yoke, and in two weights—medium and extra heavy patterns for working pressures up to 200 and 300 pounds.

"MOST supply houses sell them—yours CAN—if they DONT or WONT—Tell US"

Our Catalog is Free, Write for a Copy

The Lunkenheimer Company

Largest Manufacturers of High-Grade Engineering Specialties in the World

General Offices and Works, **Cincinnati, Ohio, U. S. A.**

Branches

New York	Boston	Chicago	London, S. E.
64-68 Fulton Street	138 High Street	186 North Dearborn Street	35 Great Dover Street

Unless You Prevent Scale Formation You Will Lose Money

Unless you *prevent* scale formation you will lose money. ¶ Scale formation does not necessarily have to cause a boiler explosion to prove that it is harmful. ¶ There are many other ways in which scale can make its presence felt to your distinct disadvantage and loss. ¶ Scale retards heat; it makes it difficult to keep up steam; it causes bagging; it means continual repairs; it means *more coal*, and consequently *more expense*. ¶ Yet if you were to have your feed water analyzed in our well equipped laboratory, a corner of which is here shown—and have Dearborn Treatment prepared for your needs— ¶ You would have clean tubes all the time, you would have no burnt tube ends, no bagging, no excessive coal bills, no night and Sunday cleaning work— You would have maximum boiler capacity, minimum expense, you would *save money*. ¶ And that's the kind of power plant service you're after, isn't it? ¶ Suppose you send us a gallon of your feed water —let our expert chemists analyze it— and we will submit you an estimate showing cost of Dearborn Treatment that will stop all scale formation in *your* boilers. ¶ This puts you under no obligations, but it will put you in possession of some figures that will surprise and please you. ¶ Send the gallon today.

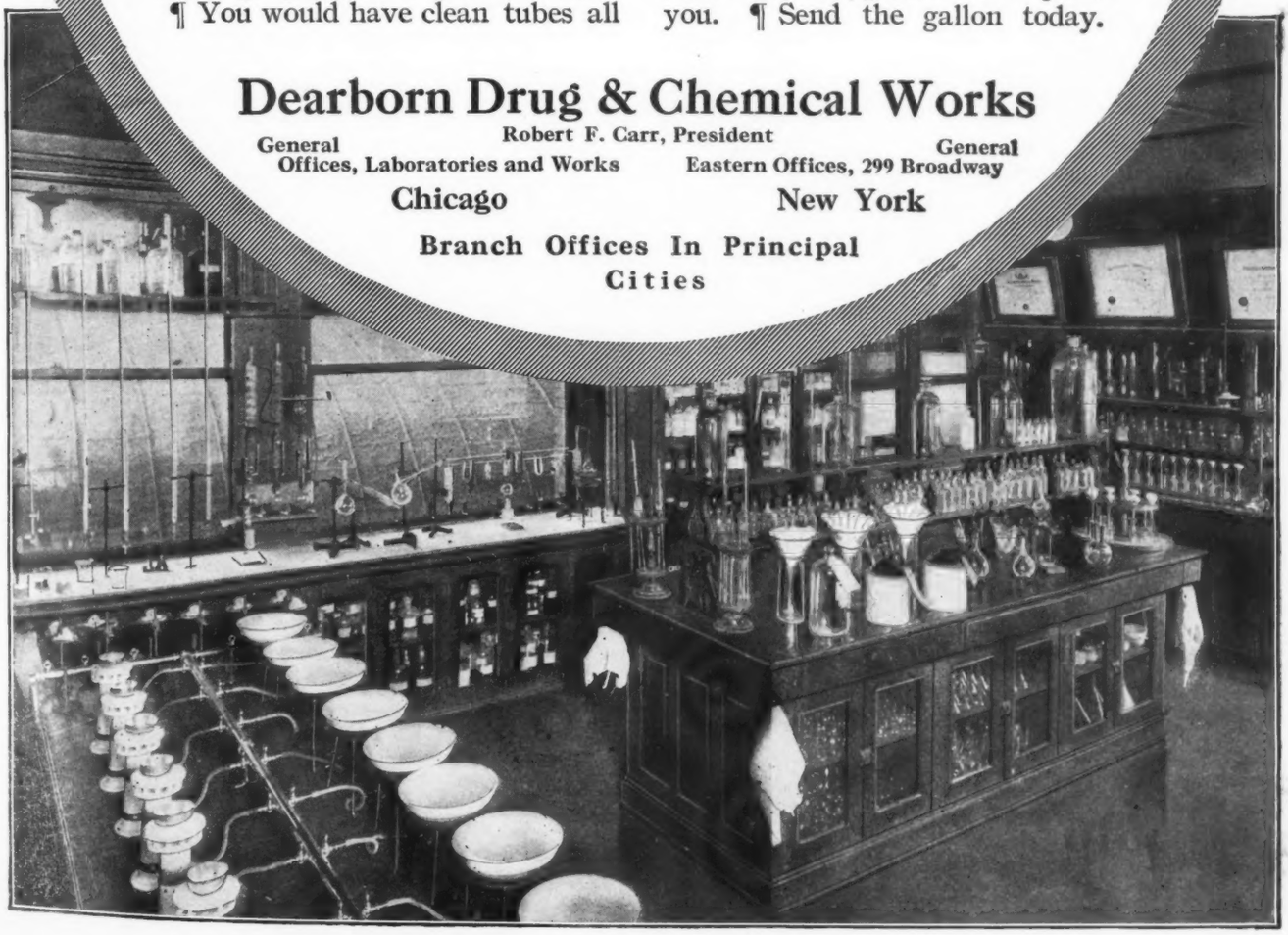
Dearborn Drug & Chemical Works

Robert F. Carr, President

General Offices, Laboratories and Works
Chicago

General Eastern Offices, 299 Broadway
New York

Branch Offices In Principal Cities

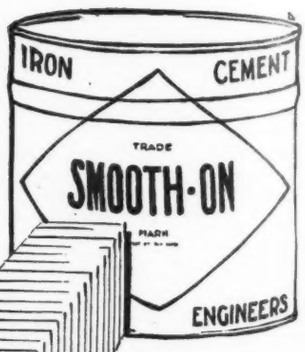


THE RIGHT WAY SEND FOR

SMOOTH-ON

INSTRUCTION BOOK

FREE



It will save YOU more time and money than anything you can have in your engine or boiler room.

It shows, by photograph and formula, how to use the various well-known Smooth-On Iron Cements in repairing boilers, engines, castings, tanks, joints, etc.

The uses of SMOOTH-ON are innumerable, and its repairs absolutely permanent.

SMOOTH-ON MANUFACTURING CO.

572-574 Communipaw Ave., Jersey City, N. J.

Chicago, 231 N. Jefferson Street **San Francisco, 94 Market Street**

London, 8 White Street, Moorfields, E. C.

How About A Strainer For Your Water Line?

Twin Strainer

You've put off installing a **Twin** long enough.

Did you ever find your water line clogged up with sticks, leaves, rocks or cinders?

If you did, you'll remember how exasperated you were, and the trouble it caused you.

That was the time you should have had a Twin in the line.

Don't get caught in that fix again.

Place your order now, and get the Strainer in service before the next high water.

We have Strainers for all sizes of plants—for all conditions of service.

Write for Catalog

Elliott Company

6908 Susquehanna Street, Pittsburgh, Pa.

An Effective Way Of Preventing Your Fuel Bills Getting The Best Of You

Cyclone Cleaner

is to keep your boilers *clean.*

There are many ways, but the *only* effective way is with **Liberty Tube Cleaners.**

There is no use in removing part of the scale and expecting good results from your boilers.

You must clean clear down to the metal to get the benefit of reduced fuel bills.

Why not give the **Liberty Cleaner** a free trial for your own satisfaction?

All you need to do is to tell us your conditions, and we'll attend to the rest.

Don't put it off another day?

Bull Dog

Liberty Manufacturing Co.

6705 Susquehanna Street Pittsburgh, Pa.

ECLIPSE SECTIONAL RAINBOW GASKET

CAN BE ADAPTED TO ANY SHAPE AND MAKE ANY AND ALL SIZES OF GASKETS.

ALL PIECES NO MATTER HOW SMALL CAN BE FORMED ON THE METAL TUBES INTO A SECTIONAL GASKET. NO WASTE OR UNSALABLE STOCK.

WILL WITHSTAND THE HIGHEST PRESSURE.

PATENTED AND MANUFACTURED EXCLUSIVELY BY

PEERLESS RUBBER MANUFACTURING COMPANY

16 WARREN ST., NEW YORK

DETROIT, MICH.:-16-24 WOODWARD AVE.
CHICAGO, ILL.:- 200-208 SO. WATER ST.
PITTSBURGH, PA.:- 425-427 FIRST AVE.
SAN FRANCISCO,-39-51 STEVENSON ST.
SPOKANE, WASH.:- RAILROAD & STEVENS STS.
SALT LAKE CITY, UTAH:-45-52 W. 2ND SOUTH ST.
ATLANTA, GA.:- 64-70 MARIETTA ST.
NEW ORLEANS, LA.,-808-821 TCHOUPITOULAS COR. JULIA ST.
LONDON E.C. ENGLAND:-11 QUEEN VICTORIA ST.
COPENHAGEN DEN:- FREDERIKSHOLMS KANAL 6
SEATTLE, WASH.:- FIRST & KING STREETS.
CHATTANOOGA, TENN.:-1106-1120 MARKET ST.
INDIANAPOLIS, IND.:- 38-42 SO. CAPITOL AVE.
DENVER, COL.:- 1556 WAZEE STREET.
HELENA, MONT.:- 113-117 MAIN ST.
PORTLAND, ORE.:- 69-75 N. 12TH ST.
PHILADELPHIA, PA.:- 19 NORTH SEVENTH ST.

FOREIGN DEPOTS

PARIS FRANCE, 76 AVE. DE LA REPUBLIQUE.
JOHANNESBURG, SOUTH AFRICA, BARSDORF B'L'D'G
PEERLESS RUBBER SELLING CO OF AUSTRALASIA, LT'D. 1 ASH STREET. SYDNEY AUSTRALIA
SYRACUSE, N.Y.-212-214 SO. CLINTON ST.
ROCHESTER, N.Y.-24 EXCHANGE ST.
ST. LOUIS, MO.,-454 PIERCE B'L'D'G.
LOS ANGELES, CAL.:- 359 NORTH MAIN ST.
BUFFALO, N.Y.:- 379-383 WASHINGTON ST.
BOSTON, MASS.:-110 FEDERAL ST.
BALTIMORE, MD.:-37 HOPKINS PLACE
LOUISVILLE, KY., NORTHEAST COR. SECOND & WASHINGTON STS.
VANCOUVER, B.C.-CARRAL & ALEXANDER STS.

THE Gasket or Joint made up from the Celebrated Garlock Style No. 900 Sheet is a guarantee of safety and economy.

Would be glad to place a *Trial Gasket* with you for *Free Test.*

Write us at once regarding this offer.

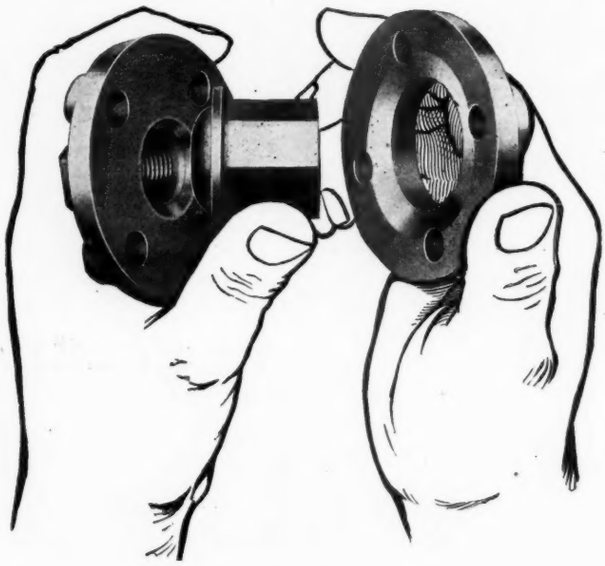
Have you a copy of our catalog?

THE GARLOCK PACKING COMPANY

MAIN OFFICES AND FACTORIES - PALMYRA, N. Y.

Branches in Principal Cities

The Jefferson 3-Part Flange Union



Makes Tight Joints That Stay Tight

A Hard Job Made Easy

¶ One of the very hardest parts of fitting flange union joints of the old style has always been to get the gaskets properly fitted and adjusted.

¶ Not only has this always been one of the hardest parts, but it has also been one of the most wasteful parts, often proving to be an operation consuming hours of time.

¶ If the pipes are out of line the gasket had to be skived because the two flange members would not present their faces in planes parallel to one another.

¶ Then again, after gaskets were finally fitted it was very probable that the bolt holes did not register opposite each other. This, in turn, meant more unscrewing and screwing to adjust the bolts—and more time wasted.

¶ But with the introduction of the Jefferson 3-Part these difficulties have been absolutely and finally overcome.

¶ For, inasmuch as the joint, with the Jefferson, is spherical it makes no difference whether pipes are absolutely in line or not.

¶ Moreover, there are no gaskets to be fitted. The joint is metallic, thoroughly ground. And the Movable Collar ends the trouble of bolt adjustment.

Fill in and return Coupon and get our new Bulletin.

Jefferson Union Co.,
Lexington, Mass

Send me a copy of your new Bulletin "A" and special information regarding the three-part Flange Union.

Name. . . . . .

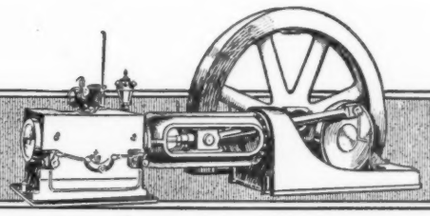
Firm . . . . . .

Address . . . . . .

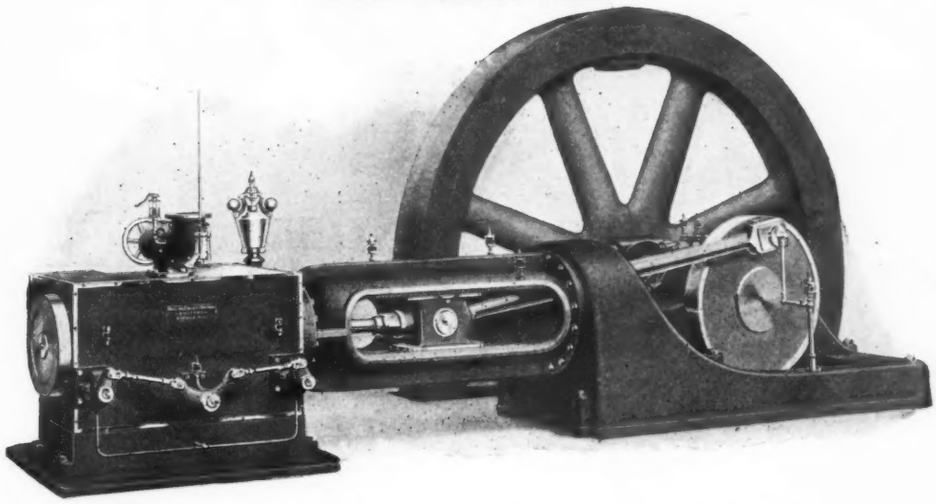
F. U.

The Jefferson Union Company

164 Main Street Lexington, Mass.



The ROLLINS ENGINE



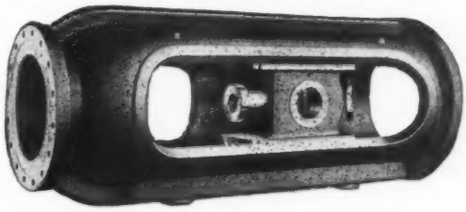
The Thing To Buy Is Engine Performance

Of course you buy an engine because of *what it will do*—and yet in most cases you don't at all.

But that's the only logical way to buy an engine of all things, because when you buy one you expect it to give you good work for twenty, thirty or even forty years.

Realizing this, we sell Rollins Engines on the "service before and after sale" idea, which means that a Rollins engine is bound to be a *profitable*, permanent investment.

In other words, our selling method means that the buyer actually buys *proved* and *guaranteed performance.*



Don't you want to buy this way?

Here's another of the series on Rollins parts—

The Rollins Heavy Duty Crosshead Guides

The trunk casting containing crosshead slides is unusually heavy, and in entire harmony with the heavy duty design.

Crosshead slides are very wide and bored in special machine that faces both ends, at same time insuring perfect mechanical accuracy. Bottom slide is water-jacketed, a feature of demonstrated merit as it prevents all distortion caused by bottom slide expanding more than the top. Keeps the slides straight and in perfect line.

Oil pockets at ends of slides provide for surplus oil. Everything about a Rollins Engine is founded on quality and nothing is dominated by cost in the least detail.

Send For A Copy Of The Rollins Engine Catalog

Rollins Engine Company, Nashua, New Hampshire

ROLLINS

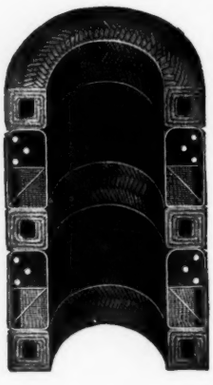
RINGS FROM OUR FAMOUS BRANDS.

STYLE 213

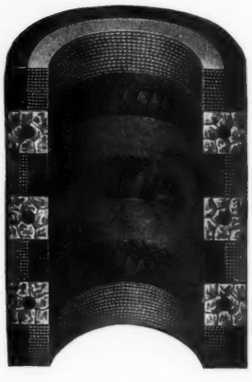
STYLE 102

STYLE 101

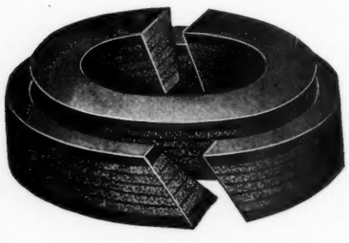
A FEW OF THOSE SHOWN IN OUR ENGINEERS' CATALOGUE. MADE TO MEET ALL CONDITIONS OF SERVICE.



STYLE 529



STYLE 528



STYLE 201

SEND FOR OUR NEW ENGINEERS' CATALOGUE

MANUFACTURED EXCLUSIVELY BY THE

NEW YORK BELTING & PACKING CO. LTD.

91-93 CHAMBERS ST. NEW YORK

CHICAGO, ILL., 130 West Lake St. ST. LOUIS, MO., 218-220 Chestnut St. PHILADELPHIA, PA., 821-823 Arch St.
SAN FRANCISCO, CAL., 129-131 First St. BOSTON, MASS., 232 Summer St. INDIANAPOLIS, IND., 120 So. Meridian St.
PORTLAND, ORE., 40 First St. SPOKANE, WASH., 133 S. Lincoln St. PITTSBURGH, PA., 420 First Ave.

LONDON, ENGLAND, 13-15 SOUTHAMPTON ROW

SATRAP

For movable bearings.

SANCHO

Sight feed and slide top.

SLIDE TOP

Plain glass oil cup.

SACHEM

For movable bearings.

SENTRY

Oiler with snap lever.

TRADE PENBERTHY MARK.

OILERS and GREASE CUPS

The "Safety," for crank pins, has a double feed and automatic shut-off. As this cup will not feed except when the engine is running, it prevents all waste of oil which would otherwise occur.

For moving bearings, eccentrics, crossheads, etc., the "Sachem," or, if a sight feed is wanted, the "Satrap," is particularly desirable because of the tight, filler stopper which makes these cups absolutely non-leakable.

The Sancho, Slide Top and Sentry are glass oilers, designed for stationary work. All these, as well as the balance of the Penberthy line, are especially designed for the work recommended. No spun brass enters their construction, every cup being made of the best, high grade, cast red brass, highly polished and guaranteed to give satisfaction in every way.

Whether or not price must be considered, you will be interested in this line, if quality is desired. Complete descriptive catalogue, prices and discounts will be sent upon request.

PENBERTHY INJECTOR CO.
355 Holden Ave., Detroit, Michigan

SAFETY

Crank pin plunger oiler.

"IXL" SPRING TOP

Self closing cover.

P. I. C.

Plain engine lubricator.

SILEX

Automatic spring compression grease cup.

SCREW TOP

Plain brass oiler with or without siphon.

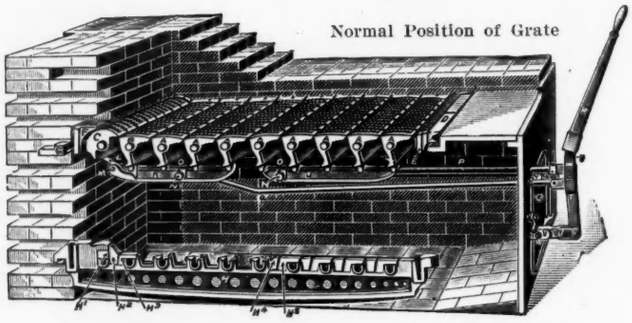
SATURN

Plain compression grease cup.

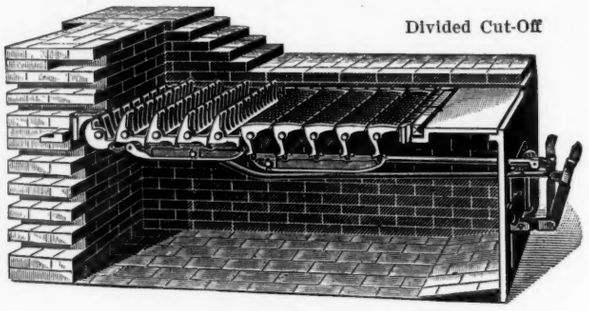
SAMSON

Screw plunger grease cup.

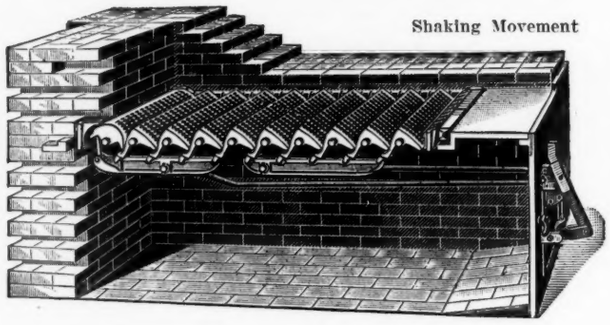
Not Only Superior Grates But A Complete System For Perfect Combustion



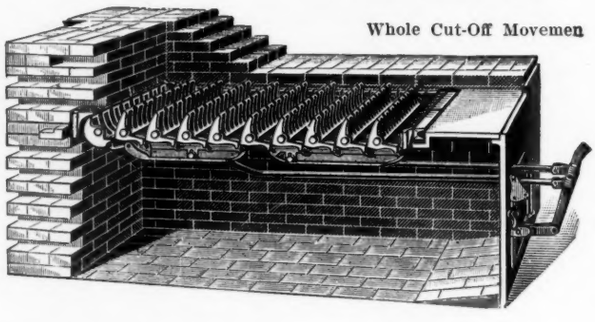
Normal Position of Grate



Divided Cut-Off



Shaking Movement



Whole Cut-Off Movemen

McClave Grates with Argand Blowers

will burn successfully all kinds of cheap small fuels, such as anthracite birdseye, bituminous slack, etc.

Note the special features of the grates as shown in the cuts. The grates alone will give excellent results in burning the larger sizes of either variety of coal by natural draft.

The Argand Steam Blower furnishes a large amount of air with a small amount of steam.

The air and the steam are thoroughly mixed in the shell of the Blower before the blast is delivered into the ash-pit.

It makes very little noise in operation and furnishes a superior quality of blast for the burning of small coal fuels.

Write for full details—ask us to send Catalog "G."

Argand Steam Blower

McClave-Brooks Co., Scranton, Pa.

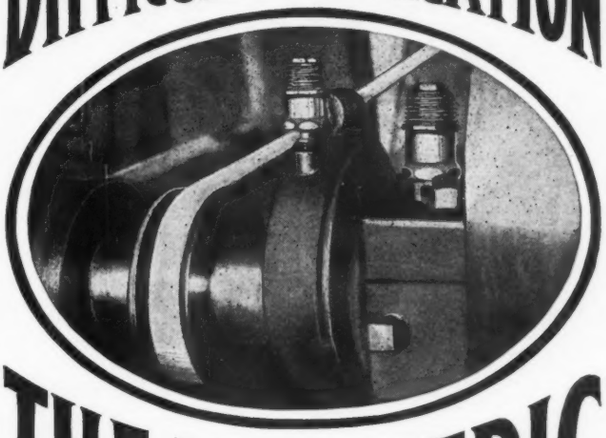
Branch Offices:

New York, 351 Fulton Bldg.
S. C. Smith, Manager.

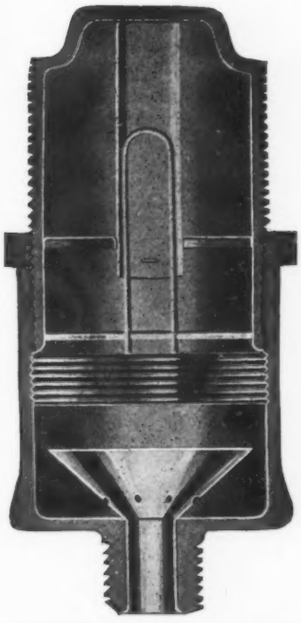
Pittsburg, 1007 Empire Bldg.
Chas. N. Hays, Sales Agent.

Chicago, 706 Fisher Bldg.
F. G. Smith, Manager.

DIFFICULT LUBRICATION



THE ECCENTRIC



PHILADELPHIA COMPRESSED AIR GREASE CUPS

The best evidence of the dependability of the Philadelphia Grease Cup for *ordinary* lubrication is its complete success on those difficult parts where other cups fail.

Probably where other cups fail most frequently are the parts where the cup is subjected to extreme vibration, centrifugal force, etc.

For instance, on your crank pin or a loose pulley; or similar cases to the eccentric shown above, where there is considerable friction between the sliding surfaces as well as centrifugal force.

Engineers have had so much difficulty in lubricating such parts as these that we find they cannot believe

The Philadelphia Compressed Air Grease Cup

can accomplish this until they see it in actual operation on just such a hard job.

And it is just for this reason that we do not ask you to accept our claim.

On the contrary, to give you an opportunity to find out before spending one cent, we will give you absolutely *free* a sample Philadelphia Cup that you can put right to work on the hardest job in your plant, and prove it for yourself.

The cup is yours to keep, whether you use any more or not.

And besides giving you a cup we will give you a free sample can of any standard grease on the market.

So the entire test, together with a free sample grease cup and grease, will cost you just a 2c. stamp.

Besides, you will save 25 to 75% of the cost of lubricating the bearing on which you test the grease.

All you need do is fill out and mail us the attached coupon, stating what size tap you desire cup for and which grease you prefer from the following:

Lubrantine, manufactured by Wm. C. Robinson & Son Co., Baltimore, Md.
Non-Fluid Oil, manufactured by N. Y. & N. J. Lubricant Co., New York, N. Y.
Philadelphia Grease, manufactured by Philadelphia Grease Mfg. Co., Philadelphia, Pa.
Zorozo Grease, manufactured by F. S. Walton Co., Philadelphia, Pa.
Albany Grease, manufactured by Adam Cook's Sons, New York, N. Y.

Send the coupon right now before you forget. It's all free.

UNIVERSAL LUBRICATOR CO.
1337 Arch Street, PHILADELPHIA, PA.

COUPON

Universal Lubricator Co., Philadelphia, Pa.
Name.....
Firm Name.....
Street Address.....
City.....
County.....
State.....
Our buyer's name is.....
State on which bearing test will be made.....
Diameter.....R. P. M.
Size of tap.....
Which grease.....
Name of dealer.....
Dealer's address.....

Power, 8-22-11

READ TH

THE

Here's Another FREE OFFER

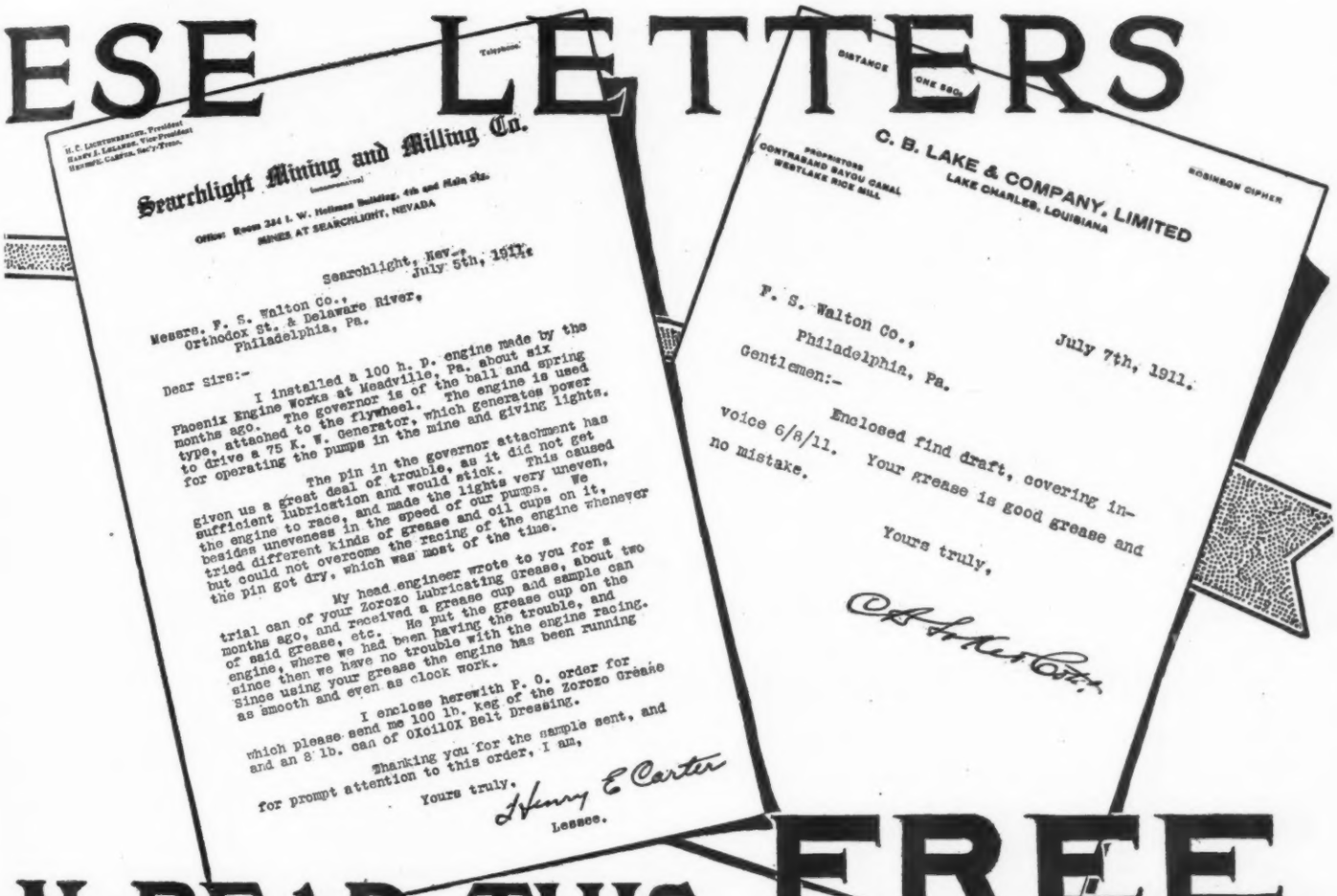
If You Prefer This One Drop Us A Line

We will send you a barrel of Zorozo and a dozen "Philadelphia" Compressed Air Grease Cups for ninety days' free trial, and if Zorozo doesn't prove better than any grease you have ever tried, you can return the cups and what is left of the barrel and no charge will be made for the amount consumed.

Or You Might Like This One Better. If So, Write Us

We will send you a trial keg and charge you at the barrel rate, payable in ninety days, and make you a present of 3 "Philadelphia" Compressed Air Grease Cups.

ESE LETTERS



Searchlight Mining and Milling Co.

Office: Room 244 I. W. Hellman Building, 4th and Main Sts.
MINES AT SEARCHLIGHT, NEVADA

Searchlight, Nev., July 5th, 1911.

Messrs. F. S. Walton Co.,
Orthodox St. & Delaware River,
Philadelphia, Pa.

Dear Sirs:-

I installed a 100 h. p. engine made by the Phoenix Engine Works at Meadville, Pa. about six months ago. The governor is of the ball and spring type, attached to the flywheel. The engine is used to drive a 75 K. W. Generator, which generates power for operating the pumps in the mine and giving lights.

The pin in the governor attachment has given us a great deal of trouble, as it did not get sufficient lubrication and would stick. This caused the engine to race, and made the lights very uneven, besides unevenness in the speed of our pumps. We tried different kinds of grease and oil cups on it, but could not overcome the racing of the engine whenever the pin got dry, which was most of the time.

My head engineer wrote to you for a trial can of your Zorozo Lubricating Grease, about two months ago, and received a grease cup and sample can of said grease, etc. He put the grease cup on the engine, where we had been having the trouble, and since then we have no trouble with the engine racing. Since using your grease the engine has been running as smooth and even as clock work.

I enclose herewith P. O. order for which please send me 100 lb. keg of the Zorozo Grease and an 8 lb. can of OXoilOX Belt Dressing.

Thanking you for the sample sent, and for prompt attention to this order, I am,

Yours truly,
Henry E Carter
Lessee.

C. B. LAKE & COMPANY, LIMITED
PROPRIETORS CONTRABAND BAYOU CANAL, WESTLAKE RICE MILL
LAKE CHARLES, LOUISIANA
ROBINSON CIPHER

F. S. Walton Co., July 7th, 1911.
Philadelphia, Pa.
Gentlemen:-

Enclosed find draft, covering invoice 6/8/11. Your grease is good grease and no mistake.

Yours truly,
C B Lake & Co Ltd

N READ THIS FREE OFFER

NO man can tell how good Zorozo Grease is until he has tried it, so we want you to try it. We have perfect faith in its ability to "make good," and are willing to stand all the expense incident to the trial. If you will do us the favor of filling in and mailing us the attached coupon, we will send you absolutely free, express charges prepaid, a can of Zorozo Grease and a "Philadelphia" Compressed Air Grease Cup to test it in, also a can of OXoilOX Belt Dressing, and a bottle of Victoria Liquid Soap—two more of the products we manufacture. An appropriate quantity of this Victoria Liquid Soap is sent with each order of Zorozo Lubricating Grease and OXoilOX Belt Dressing.

Zorozo and OXoilOX are for sale by many jobbers, but if yours does not carry them send us his name and we'll see that he does.

F. S. WALTON CO.
PHILADELPHIA, PA.

THESE FOUR SAMPLES FREE

COUPON

Gentlemen:—You may send me free Zorozo, OXoilOX, Victoria Liquid Soap and Grease Cup.

Name.....
Name of firm.....
Address in full.....
H. P. of engine.....
On what bearing will sample be tested?.....
Its diameter.....
Size of tap for cup.....
R. P. M. .....

Power-S-22-11

Boiler and Engine Room Ventilation

Is Well Nigh Perfect Where

BURT

Combination Skylights and Ventilators are Used

Here's a letter we have just received that has a bearing on the important subject of boiler and engine room ventilation.

Blytheville, Ark., Dec. 14, '08.

The Burt Mfg. Co., Akron, O.

GENTLEMEN: Referring to the 4—36" Burt Combination Skylights and Ventilators that we purchased of you some time ago, it gives us pleasure to advise you that these have given us entire satisfaction and are meeting our requirements for the purposes for which they were installed. Two of these we are using in our engine room, where they have eliminated all condensation, and are giving us perfect ventilation. The other two we are using in our boiler house, and find that they have relieved a rather uncomfortable condition that existed before these ventilators were installed. We can cheerfully recommend them to anyone in need of appliances of that nature.

Yours truly, Chicago Mill & Lumber Co.

Recently sent two repeat orders.

We have many testimonials like the above from concerns all over the country. They approve of Burt Ventilators because they are strong, extra heavy, weather-proof and they positively draw all impure air, smoke, steam, gas, etc., out of buildings with no expense whatever for operation. Study the question.

Send for our new 112-page Catalog.

THE BURT MFG. CO., 232 Main St., Akron, O., U.S.A.

Largest Manufacturers of Oil Filters and Exhaust Heads in the World.

Geo. W. Reed & Co., Montreal, Sole Manufacturers of "Burt" Ventilators for Canada.

Metal top with sliding-sleeve damper (patented).